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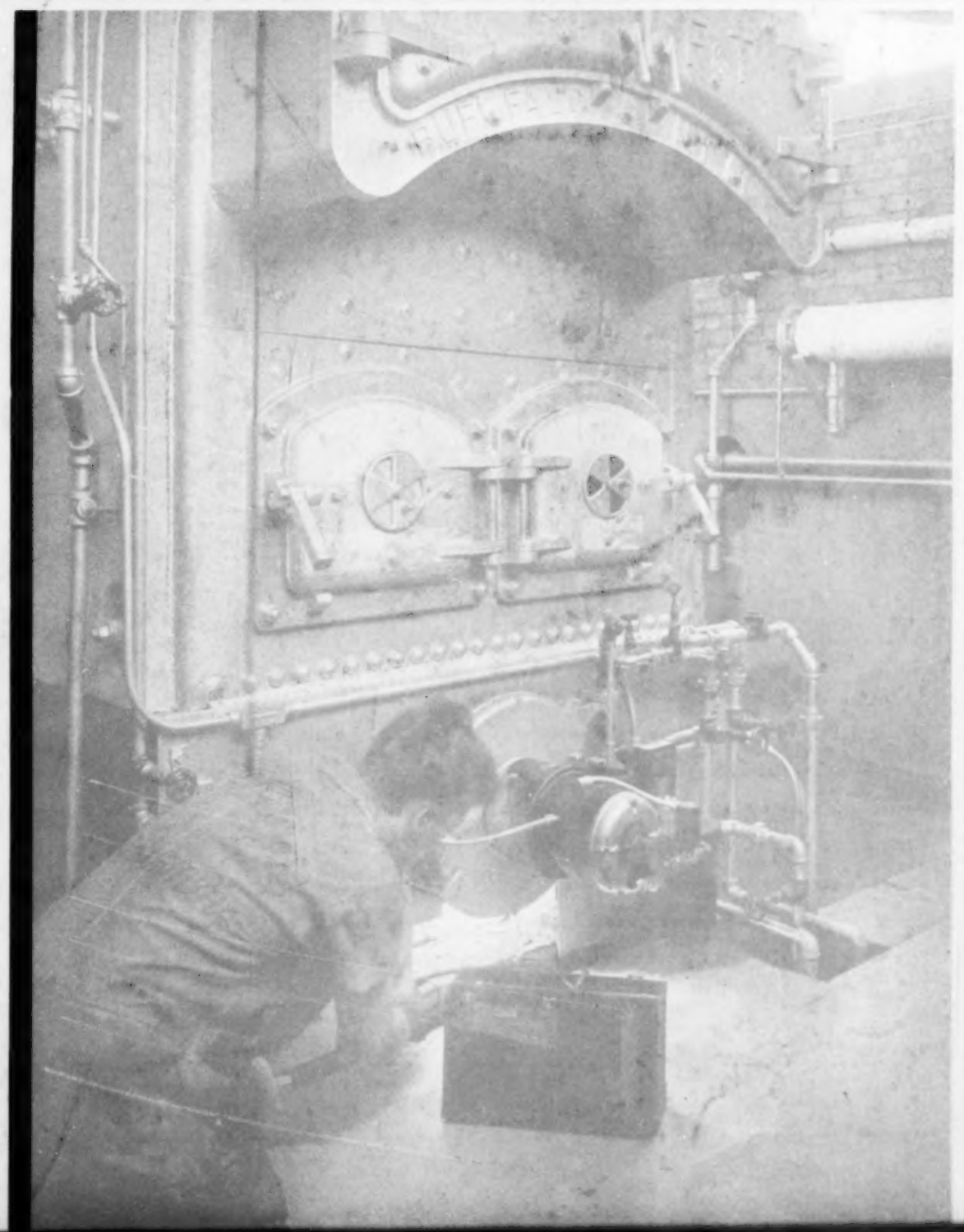

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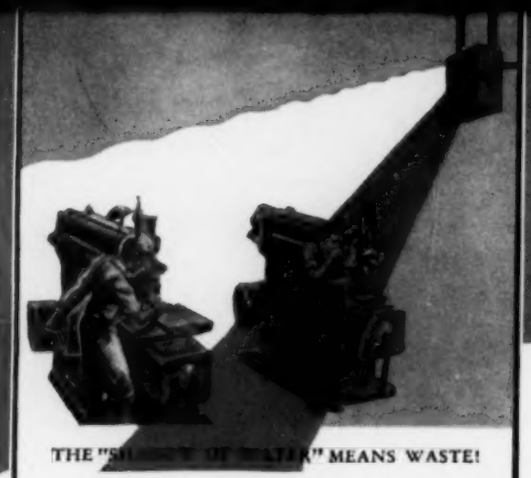
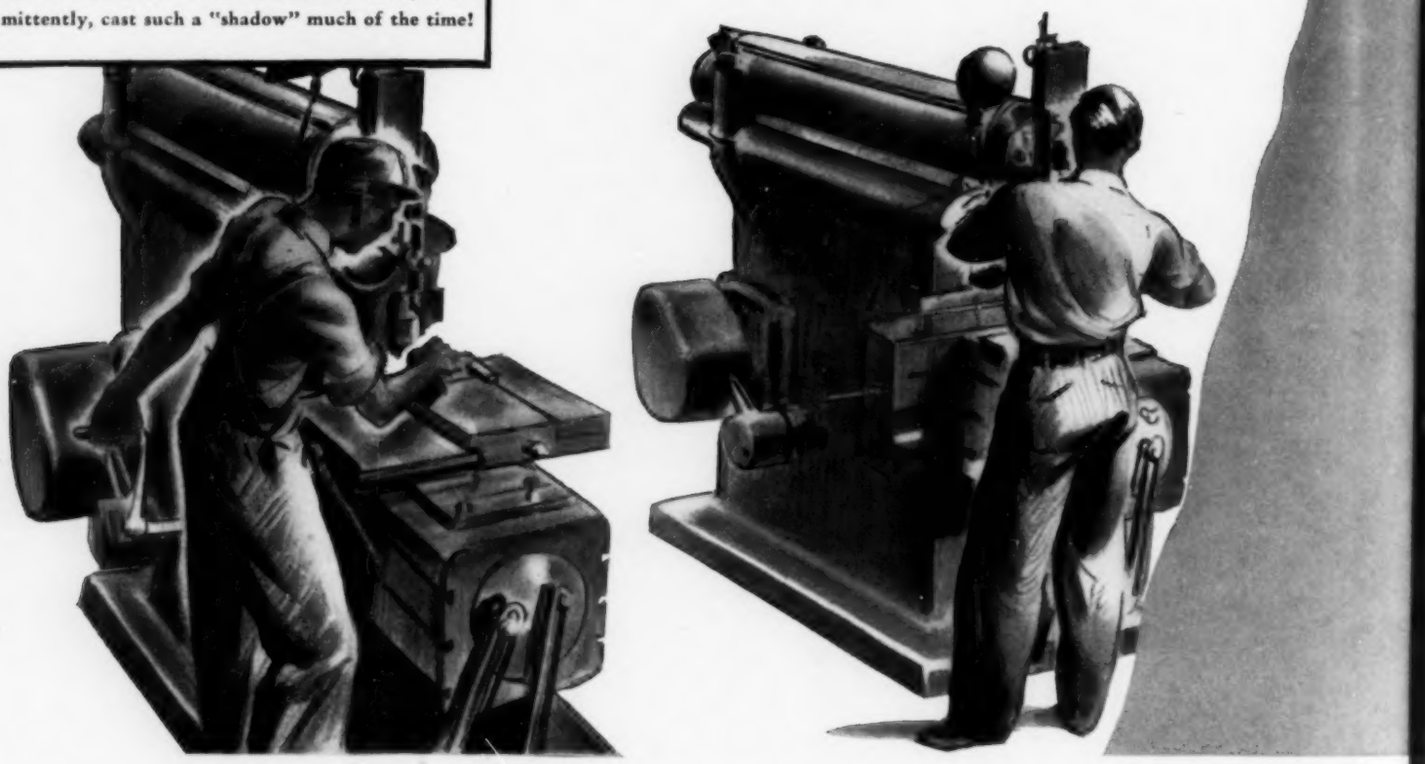


THE "SHADOW OF WATER" MEANS WASTE!

Condensed water, undrained from a unit heater, throws a long, cold "shadow" where there should be heat! Heaters that drain off condensate only intermittently, cast such a "shadow" much of the time!



NO *"COLD SHADOWS"* WITH THERMOLIER . . .

Only Thermolier offers you the exclusive Internal Cooling Leg that *continuously* drains condensed water . . . that keeps a full beam of shadowless heat pouring from all of the heating surface *all* of the time. Is it any wonder that Thermoliers deliver more heat for the money, resulting in lower fuel costs, elimination of "cold spots", better work?

Grinnell's 50 years' heating experience has built *14* outstanding points of superiority in Thermoliers! Three types and thirty sizes mean there's a Thermolier exactly fitted to your needs. Write today for the 48-page Thermolier Data Book and get full details. Grinnell Co., Inc., Executive Offices, Providence, R. I. Branch offices in principal cities.

THERMOLIER'S "SHADOWLESS" HEAT
FUEL SAVINGS LIKE

A well-known plant
savings of 29% in steam
sumption since modernizing
Thermoliers.

A railway and dock shop
replacing with Thermoliers
75 tons of coal less per week
a saving of 35%.

In just one plant of fifty
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largest manufacturers, u
with Thermolier are $50 per

GRINNELL THERMOLIER

THE UNIT HEATER WITH 14 POINTS OF SUPERIORITY

The Editor's Page

■ **Results of an** annual survey of college graduate employment prospects were made public last month by King Merritt, vice president of Investors Syndicate, Minneapolis. According to the survey, three-fourths of the June graduates from 193 American colleges, universities, technical and normal schools will be in permanent positions before the summer ends and graduates this year face the brightest job prospects in two years. . . . The survey indicated that selling is being neglected by institutions of higher learning and that it offers a stepping stone to top corporate executive positions. "Notable absentees" from the industries which visit university and college campuses to recruit new blood were amusements (excluding radio), air conditioning, hardware, jewelry, shipping and ship building, textiles (excluding specifically men's clothes and women's hose), and television.

■ **William J. Olvany** was elected president of the Heating, Piping and Air Conditioning Contractors National Association at the 50th annual convention in Chicago last month. J. E. McNevin was named vice president, and S. Austin Pope was elected treasurer. The directors are William H. Driscoll, A. F. Nass, Thomas O'Callaghan, Mr. Pope, George J. Stoehr, Daniel Hayes, Louis F. Hudepohl, and George P. Nachman. . . . Among the reports given at the sessions was that of the committee on standards, presented by Mr. Nachman, in which several revisions in the new American Standard Safety Code for Mechanical Refrigeration were proposed, as the committee feels the code contains many stringent regulations which will greatly increase the cost of air conditioning installations. In the report of the committee on air conditioning, presented by J. Lawrence DeNeille, it was recommended that market research on the fundamental question "How is Air Conditioning Sold?" be made part of the program of the committee for the coming year. . . . A comprehensive exhibit of air conditioning equipment was held in connection with the convention. The booth of the Washburne Trade School, Chicago, attracted considerable attention at the exhibit and during the show awards to the winners of a recent air conditioning contest for steamfitter apprentices at the school, sponsored by the Trane Co., were made.

■ **Air conditioning** is to be used by the Woodward Iron Co., Woodward, Ala., to produce better pig iron through controlling the amount of moisture in the air in an effort to obtain greater uniformity, according to Carrier Corp. . . . The installation for the air conditioned blast furnace requires the handling of 2700 tons of air per day, which will be cooled to a constant predetermined dew point and then heated to a temperature of 1000 F.

■ **Expenditures of the** Works Progress Administration and of sponsors of WPA projects for heating and ventilating equipment through March 31, 1939, total $9,127,014, according to the WPA division of statistics. $53,249,758 was spent for cast iron pipe and fittings. . . . The total for the value of all material, supplies, and equipment purchased for WPA work projects was $992,999,486, as of March 31. . . . Because of new federal legislation which went into effect July 1, the FHA insured modernization loan plan (Title 1) is continued for the next two years. FHA will continue to insure modernization loans made by qualified lending institutions, but the new amendments to Title 1 of the National Housing Act change several details of the FHA plan. . . . The top loan is now $2500 and if the loan is to be used for alterations or repairs, the longest term is three years. Eligible commercial and miscellaneous equipment includes built-in air conditioning systems, ash removal equipment when part of the heating system, dust collectors, and sprinkler systems.

■ **The Smoke Prevention** Association has elected Reinhard E. Kunz, chief smoke inspector of Milwaukee, its president. George G. Ritchie, fuel service engineer of the Chesapeake & Ohio Railway and John Binship, Pennsylvania Railroad, were elected vice presidents, and Frank A. Chambers, chief smoke inspector of the City of Chicago, was re-elected secretary-treasurer. . . . A meeting of the association's standards committee was held last month and it was decided that the association would confine its efforts to attempting to establish basic standards on four factors within the next few years—standards for dust measurement and concentration, minimum combustion space requirements, determining smoke density, and boiler room space.

■ **Richard P. Brown,** former president and now chairman of the board of the Brown Instrument Co., and vice president of Minneapolis-Honeywell Regulator Co., was recently appointed by Governor Arthur H. James to be the first secretary of the new department of commerce of the commonwealth of Pennsylvania. Mr. Brown has been interested for some years in the industrial development of Pennsylvania and the new department of commerce is not only to aid the state's industries in their expansion and development but is also authorized to recommend future legislation to the legislature which can be of help to Pennsylvania industries. . . . W. D. Keefe, who has been with the Fedders Mfg. Co. for the past 17 years, has taken over the duties of H. E. Rieckelman, former vice president in charge of sales of refrigeration and air conditioning products, who has resigned. . . . Allison F. H. Scott recently became associated with the Hoffman Specialty Co. in the capacity of vice president and general sales manager. Prior to joining Hoffman, Mr. Scott was an official of American Radiator Co. and later was assistant to the president of Anthracite Industries, Inc.

■ **Each of the seven** winners of the Jenkins Bros. "75th anniversary veteran valve contest" will be given a trip for two persons with all expenses paid for six days. With the exception of the metropolitan New York area winner, who will enjoy a Canadian trip, the other winners will visit New York City and the world's fair. . . . The contest was presented by Jenkins Bros. to their 500 distributors and called for a search by them and their salesmen for the outstanding "veteran" valve installation in their area, from the standpoint of length and severity of service, and cost of maintenance.

■ **The V. D. Anderson Co.,** Cleveland, Ohio, recently moved into a new two story office building fronting on W. 96th St. The building is of fireproof construction, completely furnished with the latest office equipment. . . . Opening of the new building marked completion of 59 years of continuous manufacturing, as the company was founded in 1880.

DYE BUILDING GETS GREATLY IMPROVED HEATING SERVICE

Webster Heating Modernization Includes Traps and Supply Valves for All Radiators

TENANTS GET BETTER HEATING

Cost of Heating Modernization Is Being Recovered Out of Steam Savings Effected

MEETS VARIED HEATING NEEDS

Dayton, O.—Difficulties formerly encountered in heating the 6-story Dye Building were overcome as the result of a complete Webster Heating Modernization Program.

In 1936, Philip H. Worman, Agent for the Dye Building, and Lee Schneble, the Building Superintendent, read reports of the increased heating comfort and lower fuel costs being achieved with Webster Systems of Steam Heating.

"Look over the heating installation in the Dye Building," they said to Webster Engineers, "and tell us what you would recommend to improve heating service."

Dye Building, Dayton, Ohio

Webster Engineers reported that the entire heating installation would have to be rebuilt. They recommended a unique heating modernization contract under which Ganger Brothers, Dayton heating firm, installed a Webster Hylo System and the latest type Webster Sylphon Traps and Webster Sylphon Packless Supply Valves for all radiators.

"With the Webster Hylo System," Mr. Worman said recently, "we are getting better heating service and the cost of the modernization is being recovered out of steam savings."

In the Dye Building, the Webster Hylo System meets the varied heating requirements of a W. T. Grant Co. department store, a firm of photo-engravers and general and professional office suites.

CUT STEAM COSTS IN N. P. & GT. N. RY. OFFICE BUILDINGS

Northern Pacific Saves $855.38 in One Month After Installing Webster Moderator System

INCLUDES UNIQUE INDICATOR

Great Northern R.R. Modernizes Heating in Identical Building After Observing Results

St. Paul, Minn.—The cost of heating the main office building of the Northern Pacific Railroad was reduced $855.38 in the first month after installation of a Webster Moderator System.

In St. Paul, the Northern Pacific and Great Northern Railroads have headquarters in identical buildings under the same roof. Each building has its own heating system. Both use metered steam.

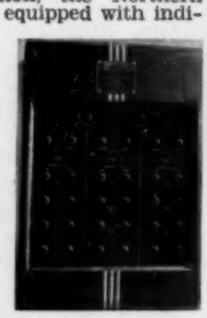
Northern Pacific Office Bldg., St. Paul, Minn.

After February, 1938, when the Webster Moderator System was placed in operation in the Northern Pacific Building, representatives of both railroads checked steam consumption figures closely to determine the effectiveness of the modernized installation.

They found that the Northern Pacific Building used substantially less steam every day than the Great Northern and secured even, comfortable heating.

Formerly, the Northern Pacific Building, because of its northern exposure, had required 15 per cent more steam than the Great Northern Building.

Before modernization, the Northern Pacific Building was equipped with individual radiator temperature control, which continues to operate in conjunction with the Webster Moderator System.



"Telltale" shows Operator that all Sections are receiving ample heat.

P. J. Leonard, of St. Paul, acted as modernization heating contractor. There is a total of 35,975 sq. ft. of installed direct radiation.

In December, 1938, a Webster Moderator System was installed in the Great Northern Building. The resulting economy and improved heating have justified the expenditure.

P. J. Leonard, heating contractor, also made the heating installation in the Great Northern Building, where there is a total of 33,283 sq. ft. of installed direct radiation.

DOCTORS' BUILDING CUTS 3-YEAR COAL COSTS $1,461

Webster Control System Saves Nearly $500 a Year in Well-Known Cincinnati Building

HEATING SERVICE IS IMPROVED

Substantial 3-Year Reduction in Heating Costs Leads Owners to Complete Payment for Cash

ADDED COMFORT FOR TENANTS

Cincinnati, O.—By preventing wasteful overheating with a Webster Hylo System of Steam Heating, the owners of the 8-story Doctors' Building saved $1,461.73 in three heating seasons.

"It seemed very unlikely that the Webster System could reduce the cost of heating nearly $500 a year in the modern, up-to-date Doctors' Building," says Morris Wasserman, Resident Manager. "But this saving was achieved, and at the same time, heating service was greatly improved."

Doctors' Building, Cincinnati, Ohio

The Doctors' Building is under the management of Theodore Mayer & Brother, of Cincinnati.

Although the owners had a contract option to pay for the heating modernization out of monthly fuel savings, they were so well satisfied that they elected to close the transaction for cash at the start of the 1937-38 heating season.

Henry Niemes, Inc., Cincinnati heating firm, acted as modernization heating contractor. There is a total of 17,700 sq. ft. of installed direct radiation.

Investigate the opportunity for profit in Webster Central Control Systems

WARREN WEBSTER & COMPANY, Camden, N. J.

Pioneers of the Vacuum System of Steam Heating . . . Representatives in 65 principal U. S. Cities—Est. 1888

Heating · Piping and Air Conditioning

with which is merged the "Aerologist"

Design, Installation
Operation, Maintenance
in Industry and Large Buildings

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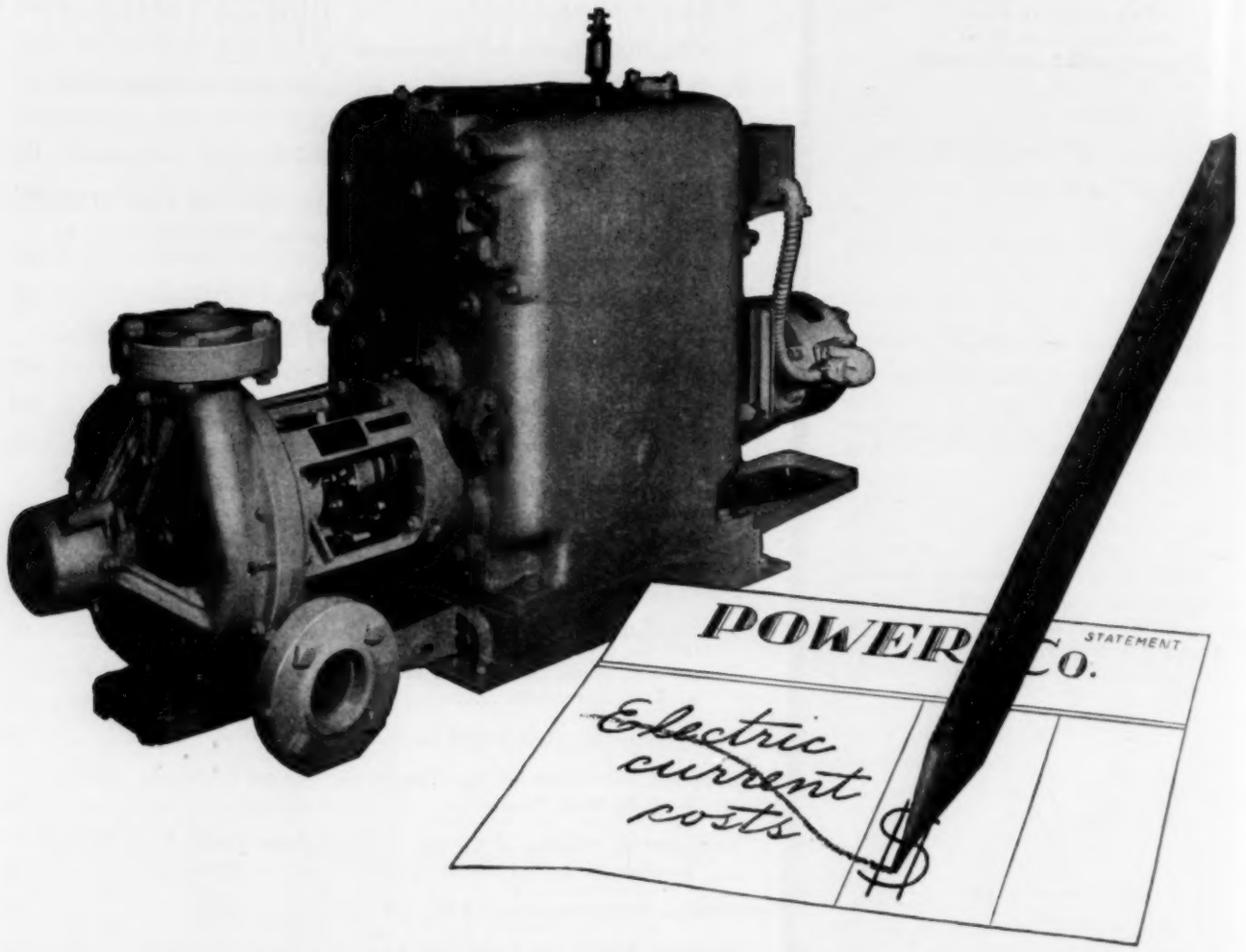
A NASH VAPOR TURBINE HEATING PUMP WILL ELIMINATE THIS EXPENSE ITEM.



The Jennings Vapor Turbine Return Line Vacuum Heating Pump requires no electric current, eliminating the one biggest expense item in the operation of an ordinary return line pump. This is because the motive power is a special turbine operating on steam direct from the heating system. Steam used to drive the turbine is returned to the system for heating, with little heat loss.

The really important saving promoted by the Jennings Vapor Turbine is in the system itself, however, for this is the only heating pump that can operate continuously with economy. Continuous operation means uniform circulation, and uniform circulation means a big saving in steam.

This pump has but one moving part, no internal wearing parts, and no internal lubrication. It is compact and quiet, and will give long and satisfactory service at the lowest maintenance cost. Bulletin No. 246 tells all about it. It is free on request.

THE NASH ENGINEERING COMPANY

SOUTH NORWALK, CONNECTICUT, U. S. A.

Heating · Piping and Air Conditioning

Vol. 11, No. 7 • *July, 1939*

Attention to Heating Plant Now Will Pay Dividends Next Winter

By Lewis W. Mauger*

FROM the moment a new building or piece of equipment is completed, a constant vigil must be maintained to assure its continued operation at peak efficiency. There are two phases to maintenance, the first relating to cleanliness and appearance, and the second applying to the mechanical operation or physical condition. With the advent of modern building management methods, which have necessarily been sharpened by the demands of competition and by the business depression, greater attention has been paid to all phases of expense and special emphasis has been given to heating. Proper attention to the heating plant during the summer season will pay dividends in better operation next winter.

Cleaning Heating Boilers

It was formerly customary to give a superficial cleaning to heating boilers before the summer shutdown by removing ashes from grates and pits and wirebrushing accessible heating surfaces. In many types of boilers there are gas passes which the usual flue brushes cannot properly reach, with the result that soot accumulates over a period of time, making these parts of the boilers useless for heat transmission. The cleaning of the water side was often considered complete when the boiler was drained, flushed and refilled. The accumulated scale, rust and dirt deposits were not removed by this method and subsequent cracking of sections or corrosion of tubes, as well as decreased efficiency, resulted.

It has been found well worthwhile, both from the standpoint of operating efficiency and increased boiler life, to pay as much attention to the cleanliness of boilers as to the appearance of the occupied portions of a building. A relatively small amount of soot or scale reduces heat transfer by a comparatively large percentage, so that dividends are paid by keeping these insulating deposits at a minimum.

If high pressure steam or compressed air is available, soot blowers should be used in connection with water tube boilers. A heavy duty vacuum cleaner can also be used to advantage, especially to clean locations difficult to reach with the available tube brushes.

In general, the soot from small boilers using anthracite coal on natural draft should be removed once every month. Boilers using soft coal on mechanical draft should be cleaned twice a month. High pressure boilers

Modern plant and building management methods—sharpened by the demands of competition and the depression years—have meant that greater attention has been paid to all phases of expense. Special emphasis has been given to the heating item. . . . Dividends in better, more efficient heating plant operation will result next winter if planned summer maintenance work is thoroughly done. Mr. Mauger offers suggestions on what to do now to keep and put heating equipment in good operating condition

*Supervising Engineer, Brown, Wheelock, Harris, Stevens, Inc., Real Estate.

Treatment of Boilers During Non-Heating Season

1) At least a month before the end of the heating season a cleaning compound should be added to the boiler water and allowed to boil through the system for a week or two, after which the system should be flushed, drained and refilled several times until the water runs clean. Water tube boilers or boilers having hand- or manholes should be opened and the tubes and drums thoroughly cleaned of any remaining sludge. A small additional portion of cleaning compound should then be added and the boiler filled to the safety valve for the summer.

2) Clean all soot and scale from sections and tubes until the metal is visible.

3) Clean all ashes from pit and grates.

4) Thoroughly clean smoke breeching of all soot and, if possible, remove and store in dry place.

5) Clean stack base.

6) Wire brush and paint all external metal on boilers with heat resisting aluminum paint. Apply light mineral or garage drain oil to all exposed metal in fire box, ashpit and gas passes.

7) Check over and repair all brick work in fire box.

8) Repair or replace all defective parts in the system, such as gages, valves, traps, asbestos covering, damper regulator, diaphragms, etc.

At the start of the following firing season

1) Lower water to normal level.

2) Replace breeching.

3) Seal all cracks and draft leaks with boiler cement.

4) Close clean-out doors and seal with asbestos cement.

Care and Upkeep of Heating Plants

1) Keep connections between boiler and gage glass clean and maintain proper water level.

2) If water level runs low daily, check returns and drips for water leaks, radiator control and all air valves for steam leaks and notify the district representative at once if you cannot overcome the condition.

3) Keep the ashpit clean. Do not force the grates if jammed; remove clinkers through firing door.

4) Allow sufficient time to get up steam in the morning to eliminate the necessity of forcing the fire and the possibility of burning grates, tubes or sections.

5) Clean the tubes regularly in horizontal return tubular boilers.

6) Burn proper size and quality of coal to fit grates and boiler design. Maintain even and medium depth of fire bed.

A few primary requisites of proper design are:

1) Feed water should be introduced into the return line from the hot water supply.

2) There should be at least a 2 ft difference in elevation between the lowest point in the steam lines and the boiler water level to maintain a quiet system and dry steam.

3) The depth of the ashpit should not be reduced below that recommended by the boiler manufacturer.

4) The heating plant should be provided with a "Hartford loop" as protection against a sudden leak in the return line.

and boilers which are forced beyond the manufacturer's rating should be cleaned weekly.

If satisfactory conditions can not be maintained by these methods, some of the compounds which will remove soot by reducing its burning temperature may occasionally be blown into the firebox. Care should be exercised in selecting these materials to be certain that they are non-explosive and non-corrosive. Furthermore, they must be applied in such a manner that backfires can not result.

In using test data to calculate the percentage loss caused by soot on fire tube boilers, correction should be made for coal fired jobs in which most of the deposits accumulate in the lower section of the tubes, whereas the hottest gases pass through the top portion of the tubes. The loss in efficiency would therefore not be in proportion to the average thickness of the soot but would be considerably lower.

Protection Against Corrosion

During the summer, corrosion can be minimized by covering exposed metal surfaces with some form of mineral oil, such as garage drain oil. Exterior exposed metal should be kept painted, heat resisting aluminum paints being very satisfactory.

The water side of boilers requires a more scientific treatment to render the feed water non-corrosive and to convert hard, scale forming ingredients to softer compounds. Sodium salts and other phosphates are popular for producing these results but their application should be supervised to maintain the boiler water in suitable condition. Too small dosages may result in the formation of scale and the corrosion of metal parts, whereas too large dosages may produce priming or caustic alkalinity. In the case of new boilers or systems in which piping changes have been made with the introduction of oil in the boiler water, special treatments are also required to remove the oil by absorption or reaction. It is often wise to blow the water from the top until all traces of oil have disappeared.

In addition to the chemical treatments which have been in use in power plants for many years, there are available colloidal solutions which remove foreign ingredients from water because of their flocculent nature and at the same time, it is claimed, protect the metal by a thin film of material.

As important as the treatment added to the water is the proper elimination of accumulated solids by means of adequate blowdown. Tests of the boiler water should be made from time to time to ascertain whether the blowdown has been sufficient. Many older boiler plants have no means of blowing down at the proper points or in sufficient volume. The lowest point of boilers, headers and return lines should be equipped with adequate sized valves so that sludge may be removed as it accumulates.

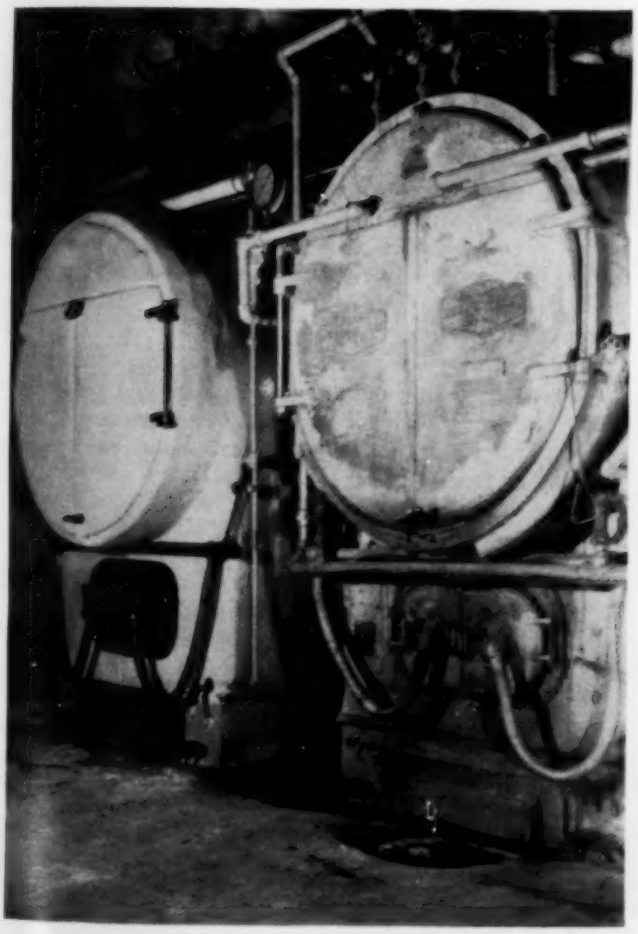
From time to time and particularly at the close of the heating season, the hand and manhole plates of steel boilers and the header caps of water tube boilers should be removed and an examination made of the interiors. If the treatment has been proper, no hard scale should be present and the soft materials may be washed out by

hosing. Should any scale be found in water tube boilers that can not be removed by a tube brush, a water or steam driven turbine will usually be effective but, due to its action on the metal, its use should be restricted as far as possible.

There has been a long-standing disagreement as to whether a steel boiler out of service should remain empty or filled with water. If a boiler remains empty, some moisture absorbing material, such as calcium chloride, should be set inside to prevent condensation of moisture on the steel surfaces with subsequent corrosion. This is also advisable for the fire sides of boilers which are left full of water. The water in a boiler which is to remain filled should also receive some treatment and should be brought to the boiling point to expel air before laying up.

In many plants hot water is generated by coils located within or adjacent to the heating boilers and heated by the boiler water. Some of these coils feed direct to the hot water system through mixing valves and others circulate to storage tanks. In both instances the coils should be equipped with blowdown facilities to keep them functioning with good efficiency. This is particularly true of coils connected to storage tanks because the coils are the lowest point in the entire hot water system and become the depository of any foreign matter present. It is not uncommon to increase the hot water temperature 40 or 50 deg by blowing down and back flushing these coils.

Plastic covering applied to boiler doors and fronts, as shown at the left, aids in increasing efficiency



Summer Inspection of System

During the summer, thorough inspections of all heating equipment should be made and repairs arranged to be completed prior to the next heating season. Particular attention should be paid to the condition of the brick work, grates, tubes, breechings and safety devices. Many building managers have standard questionnaire forms which are used annually to form a basis for repair and improvement schedules. A form used by one of the larger management offices in New York City for coal fired heating plants is shown herewith. An instruction form covering the treatment of boilers during the non-heating season and the care and upkeep of heating plants during the heating season is also shown.

The summer is the proper time to test some of the automatic equipment used in the heating system and to effect improvements to accelerate circulation or reduce costs. The drip traps on the vacuum steam heating systems should be tested annually by removing the thermostatic elements and trying them in a test body. The elements of all radiator traps should be similarly tested about every three years and those which are defective should be replaced. Vacuum pumps should be tested to determine if sufficient vacuum can be pulled, and defective pumps should be overhauled.

Systems of automatic control or temperature regulation need cleaning and checking, as dirt or corrosion may have affected their action. Ducts in warm air or indirect heating systems should be examined each summer and vacuum cleaned when necessary. Accumulations of dirt are often found on indirect heating stacks, reducing appreciably the heat transfer and air flow.

Maintenance of Firing Equipment

Stokers should be thoroughly cleaned at the end of each season and the worm and other moving parts exposed for inspection. Defective tuyeres, fans or controls should be replaced. Where stokers have been improperly set without allowance for sufficient combustion space between the level of the fire and the heating surfaces, improved efficiency can often be obtained by resetting them to lower levels or by raising the boilers to provide adequate combustion volume.

Oil burners and appurtenances should be thoroughly cleaned to remove carbon and sludge. In the past it has been the practice to pump out oil storage tanks and to scrape out sludge every few years. There are now on the market a number of compounds which will dissolve sludge and dissipate water, allowing both of these impurities to pass through the suction lines without adversely affecting the burner operation. In some instances, improvements in boiler efficiency and reduction of soot and smoke have also resulted from the use of these materials.

Controls and Insulation

Safety devices for mechanically fired boilers often receive little or no attention because they are called upon to function so infrequently and their importance is often discounted. Several instances have come to the writer's

attention during the past few years in which low water cutoff devices have not been properly cleaned and adjusted and have remained inoperative when boiler water has failed to return, resulting in the burning out of tubes and cracking of sections. Stack switches, protective relays, aquastats and other controls should receive their share of attention.

Losses from exposed hot metal surfaces can be minimized by replacing defective covering on boilers, hot water tanks, piping, etc., and increased efficiency can be obtained by insulating parts of boilers that it has not been customary to cover. These parts include boiler fronts, clean-out doors and brick settings. Some materials are now available which may be applied directly on the metal or brick surfaces, but best results are usually obtained if they are reinforced with studs and wire.

For buildings using high pressure or purchased steam, it is particularly important to minimize losses due to leaks or lack of covering. All control valves, reducing valves, regulators and traps through which steam may blow should be tested frequently and discs or seats renewed as soon as leaking is apparent. Because of the possible corrosive action of oxygen and carbon dioxide in the steam condensate, an effort should be made to keep air out of vacuum systems by using only vacuum type air eliminators and sealing all leaks. Return lines in some buildings have corroded through or clogged due to this action of the returning water.

During the heating season it is advisable to test boiler efficiencies from time to time by taking instrument readings of the draft, CO_2 and flue gas temperatures. These three indices not only provide data for calculating actual efficiencies but often point to the means of correcting faults. Recent tests of a large group of boilers showed that a surprising number were operating around 50 per cent boiler efficiency which comparative minor changes such as increasing the thickness of the fuel bed, controlling the draft, or adjusting the air supply corrected. If good efficiencies are to be maintained, these tests should be repeated monthly at first and then with less frequency as the operating personnel becomes accustomed to the proper methods of control.

Annual Heating Report

BUILDING..... SUPERINTENDENT..... DATE.....1939

Please answer the following questions where they apply to your building. Our *coal records,* also our summer repair schedule, depends on the *prompt* reply from *all* our buildings and your cooperation is requested to eliminate delays and the necessity for additional reminders.

1) What percentage was your building occupied during the past season?.....% What percentage was your building occupied during the previous season?.....%

2) Is hot water supplied?.....How is it heated? .....(by individual heater, submerged coils, etc.)

3) How many heating boilers are there in your building? .....They were manufactured by the.....Co. and are of the water tube ( ), fire tube ( ), sectional ( ) type. (Check type of boiler.)

4) The bunkers have a capacity of.....tons. (A ton occupies 35 to 36 cu ft)

5) From October 1, 1938, to May 31, 1939.....tons of.....size coal were burned for heat and..... tons of.....size coal were burned for hot water. This was delivered by the.....Coal Co.

6) The *total* consumption of coal during June, July, August and September, 1938, was.....tons of.....size coal.

7) On May 31, 1939, there were how many tons of the various sizes of coal on hand and how many more tons of the various sizes did you have room for?

Size of Coal	*On Hand (tons)*	*Room for (tons)*

8) Do you need any grate bars?.....How many?.....

9) Does water leave the boiler(s) at any time?..... Does all condensation return to the boiler(s)?.....

10) Are any repairs necessary to the boiler brick work? .....To what extent?.....

11) Has the inside (water side) of the boiler(s) been properly cleaned?.....On what date?.....

12) Have the smoke pipes and connections been thoroughly cleaned?.....On what date?.....

13) Have the tubes, heating surfaces, ashpit, etc., of the boiler(s) been properly cleaned?.....On what date? .....

14) Will you need a contractor to take care of item 10 ( ), item 11 ( ), item 12 ( ), item 13 ( )? Answer each with "Yes" or "No."

15) Have you any suggestions for improving the operating efficiency of the heating plant? (Traps, valves, vents, covering, changes in draft control, method of firing or grade of coal, etc.) Please explain.

16) What improvements in equipment or operation were made since May, 1938?

17) Is there any heating equipment in your building (such as boilers, blowers, dampers, vacuum pumps, etc.) which is not being used and could be removed?.....In what condition is it?.....

Steam Circulation

The circulation of steam to various sections of a building should be checked occasionally to determine the lowest operating pressure at which steam can be supplied to all parts and to find the time required to accomplish full circulation. If different pressures are required for various parts of a building, the cause should be found and corrected if possible. In most instances, circulation can be accelerated by properly pitching or dripping trapped lines, resetting radiators and, in the case of steam pressure systems, adding quick vents at strategic points to allow rapid removal of air. In vacuum systems, if it is difficult to retain sub-atmospheric pressures, the elimination of leaks will improve the even supply of steam to the various parts. Leaks are most often found in expansion joints, radiator connections and valve bonnets. In general, higher vacuums mean improved circulation with lower fuel costs and attention paid to improving vacuum will be well repaid.

General Maintenance Rules

The maintenance of heating systems may be summed up by the following general rules:

1) Watch the fire side of boilers and remove deposits as they occur.

2) Test the water to maintain proper alkalinity and softness and blow down frequently.

3) Operate on as low a pressure and as high a vacuum as is possible and, when it becomes necessary to increase pressure or when vacuum decreases, search for the cause and correct it immediately.

4) Automatic controls, whether for draft, pressures, zoning or fuel oil, should be kept in automatic operation. These devices will function properly only so long as they are given intelligent attention.

Steels for High Pressure and Temperature Service

By J. C. Hodge*

THE relationship between the short time tensile strength of a 0.20 per cent carbon steel, having at room temperature a minimum tensile strength of 60,000 lb, and its allowable working stresses throughout a range of temperature given by two pressure vessel construction codes, is shown in Fig. 1. Below 650 F, design is based on the same allowable working stress, established as a certain percentage of the room temperature tensile strength. Above this temperature permissible stresses decrease, reaching very low values at 950 to 1000 F and suggesting the desirability of higher strength steels for this temperature range. Above this temperature, oxidation of the metal becomes an important factor, necessitating the use of alloy steels of superior resistance to oxidation. The subject of high pressure, high temperature metallurgy may therefore be conveniently divided into:

1) High pressures at temperatures below approximately 650 F, where elastic properties of the steel are of primary importance.

2) High temperatures between 650 F and 950 to 1000 F, where high temperature strength is important but where oxidation is not a factor.

3) High temperatures above approximately 950 to 1000 F, where oxidation is a factor.

High pressure, high temperature metallurgy requires for the application of any steel the following considerations:

1) Elastic properties at normal relatively low temperatures.

2) Strength at elevated temperatures, as related to its short time tensile strength and to the phenomenon of creep or slow plastic deformation under stress.

3) Chemical stability to environment, such as resistance to oxidation by steam, air and furnace gases; resistance to other corrosive influences, such as intergranular embrittlement by caustic soda or by hydrogen.

4) Structural stability or absence of microstructural changes whereby inferior physical properties are developed after heating at service temperatures.

5) Stress gradients as a result of temperature gradients and their magnitude as related to the coefficient of thermal expansion and the modulus of elasticity of steel.

6) Fabrication problems, including the important factor of weldability of the steel.

7) Economy of use; i. e. service performance as related to initial cost.

Since the allowable working stress of a steel for temperatures below approximately 650 to 700 F is based on the room temperature tensile strength (or in some cases the yield strength), economy in weight of pressure vessel or drum may be attained by using steels of higher strength than the usual carbon steel of 0.30 per cent carbon maximum and of 55,000 or 60,000 lb per sq in. minimum tensility. Increased strength is attained by the addition of alloying elements, the individual elements effecting increased strength to different degrees (Fig. 2). Note that increased yield strength is obtained in all cases but that some of the elements, notably copper, phosphorus, nickel and silicon, are particularly effective in increasing the yield strength without unduly increasing the tensile strength.

One of the four low alloy, high strength steels recognized by the ASME boiler code is a modified carbon steel with increased manganese and having a minimum tensile strength of 65,000 (Grade A) or 70,000 lb per sq in. (Grade B). It has been widely used for pressure vessel and boiler drum construction, large numbers of fusion welded boiler drums of this steel having been fabricated since 1934, to the practical exclusion of 55,000 lb tensile boiler plate—at least for the heavier drums. The most interesting example of this modified carbon high strength steel lies in its use for the Boulder Dam penstock piping, where design requirements were based on a 2:1 ratio of yield strength to working stress, the nominal working stress of 19,000 lb per sq in. necessitating a steel of 38,000 lb per sq in. minimum yield strength. A proportionate saving in weight of structure accrued from its use over the use of lower carbon boiler plate.

Since this modified carbon, high strength steel is inexpensive relative to steel of any appreciable special alloy content, other low alloy steels of approximately the same tensile strength will find only restricted use in the relatively low temperature range of the high pressure field; i.e. for special purposes, where a special property of a particular alloy steel is necessary, as for example the 2½ per cent nickel steel with its notch toughness at sub-zero temperatures.

While these low alloy steels may be produced as boiler plate with tensile strengths of the order of 90,000 to 100,000 lb per sq in., fabrication hazards in the construction of heavy pressure vessels by fusion welding militate against the use of such high strength steel. The ductility of very high strength fusion weld metal has also been lower than desirable for important service and will require improvement for further upward extension of tensile strength of pressure vessel steels.

In a talk on high pressure, high temperature metallurgy at the Midwest Power Conference, Mr. Hodge discussed the requirements for ferrous metals and alloys for such service. . . . He subdivides the subject into high pressures at temperatures below approximately 650 F, where elastic properties of the steel are of primary importance; high temperatures between 650 and 950 to 1000 F, where high temperature strength is important but where oxidation is not a factor; and temperatures above 950 to 1000 F, where the oxidation factor must be considered

*Chief Metallurgist, Babcock and Wilcox Co.

Fig. 1—Relationship between short time tensile strength and allowable working stresses of 0.20 per cent carbon steel (minimum tensile strength 60,000 lb) . . . . . Fig. 2—Effect of adding various elements on yield strength and tensile strength of iron (Epstein, Nead and Halle) . . . . . Fig. 3—Allowable working stresses (ASME code) of carbon-molybdenum steel, 2½ per cent nickel steel and carbon steels . . . . . Fig. 4—Experimental values for the oxidation of iron and mild steel in air. (British Scientific Industrial Research—"Oxidation and Scaling of Heated Solid Metals") . . . . . Fig. 5—Relative oxidation resistance of chromium steels and irons . . . . . Fig. 6—Effect of silicon on the oxidation resistance of low carbon silicon transformer steels in air atmospheres

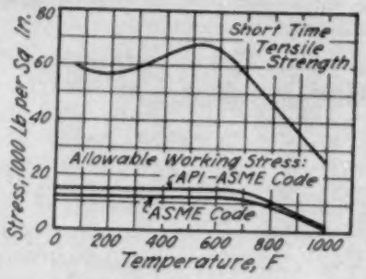


Fig. 1

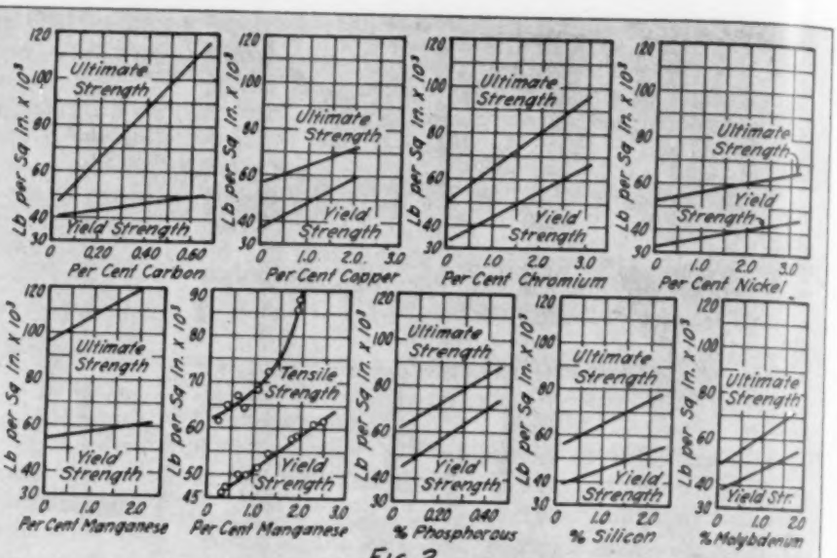


Fig. 2

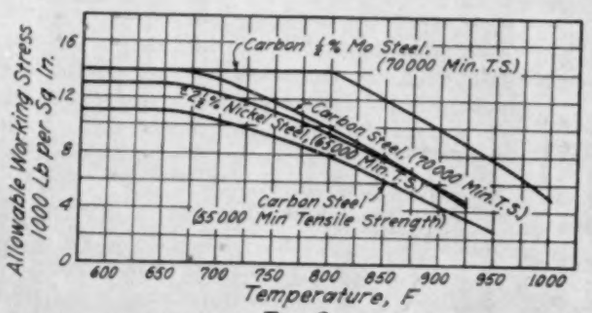

Carbon ½% Mo Steel, (70000 Min. T.S.)
2½% Nickel Steel, (85000 Min. T.S.)
Carbon Steel (70000 Min. T.S.)
Carbon Steel (55000 Min Tensile Strength)
Allowable Working Stress 1000 Lb per Sq In.
Temperature, F

Fig. 3

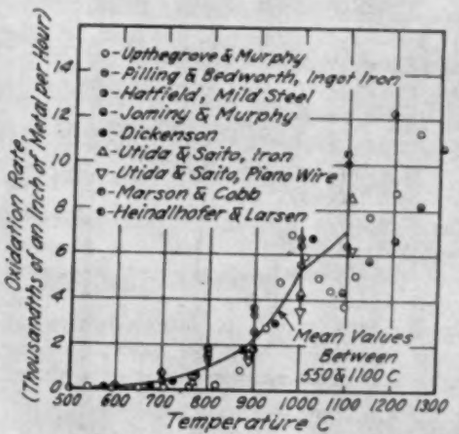

Upthegrove & Murphy
Pilling & Bedworth, Ingot Iron
Hatfield, Mild Steel
Jominy & Murphy
Dickenson
Utida & Saito, Iron
Utida & Saito, Piano Wire
Marson & Cobb
Heindlhofer & Larsen
Mean Values Between 550 & 1100 C
Oxidation Rate, (Thousandths of an Inch of Metal per Hour)
Temperature C

Fig. 4

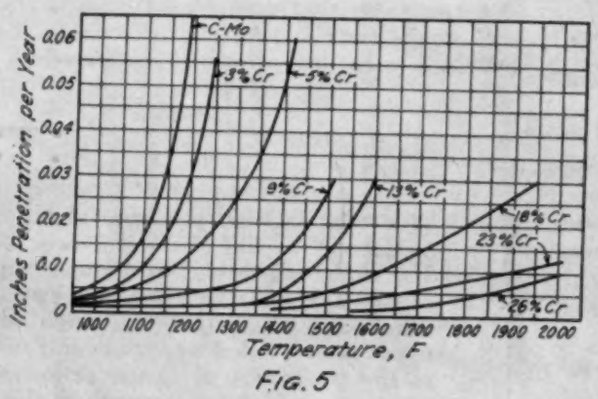


Fig. 5

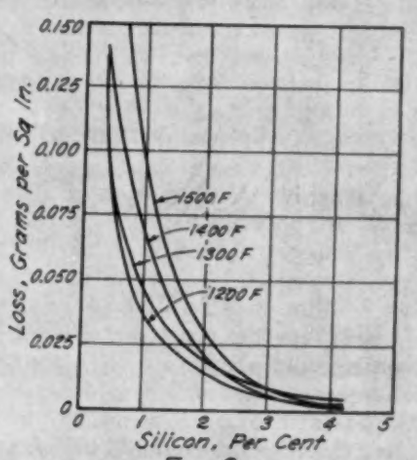


Fig. 6

In the 650-950 F temperature range oxidation is not a factor, and the desideratum is a steel of increased resistance to creep. Economy dictates the use of a high carbon steel (0.35 per cent max) for the low end of this range, where elastic properties are still predominant and where this carbon steel possesses working stresses but slightly less than those of carbon molybdenum steel (Fig. 3). At temperatures between approximately 850 and 950 F, however, steel with ½ per cent molybdenum possesses a materially increased strength over plain carbon steel, permits proportionately higher design working stresses, and may be used to economical advantage.

The ½ per cent molybdenum steels have excellent weldability, provided the carbon content is low. No serious difficulties have been encountered in welding a number of heavy walled pressure vessels for the oil industry. A drum with a wall thickness of 4⅛ in. to operate at 975 F is now under construction. It has been widely used for superheater tubular elements and headers and for power plant connecting piping from the superheater outlet to the turbine. These lines generally involve cast valves of ½ per cent molybdenum but with carbon content as high as 0.30 per cent, necessitating preheating in welding to a valve. Preheating for carbon contents in excess of approximately 0.15 per cent is generally recommended; the higher the carbon content, the greater the preheating temperature employed.

Above a temperature of approximately 950 to 1000 F, plain carbon steel shows increasing oxidation or scaling with increasing temperature. The mechanism of scaling of iron, steel and other metals by various oxidizing agents is fairly well understood and in the case of steel in contact with air and oxygen the reaction rate has been fairly definitely established through a wide range of temperatures. Fig. 4, reproduced from a recent British publication, presents the oxidation rates of iron or mild steel by air, as obtained by reducing the values of various investigations to a common basis, assuming a parabolic relationship between rate of scaling and time. Scaling proceeds by diffusion of oxygen through any oxide formed, and since, for steels, the oxide formed is nonporous and in a state of compression from the volume expansion attending its formation, the oxide resists diffusion of the oxygen to the underlying metal so that the rate of oxidation falls off with time. Increased protection is, therefore, obtained with time, the oxidation rate following a parabolic equation.

Some deviation from the mathematical expression of the oxidation rate may be expected for a number of reasons, the chief being a tendency to cracking of the oxide film. Cracking and peeling of the oxide scale may be produced by alternate heating and cooling. The tendency to cracking will be greater as the scale becomes thicker.

Increased resistance to oxidation may be imparted to steel by the addition of certain alloying elements, such as chromium, silicon, aluminum, titanium or columbium, the use of chromium being most common. These elements tend to produce tight surface layers of oxide, which retard oxygen diffusion and protect the underlying metal from attack. Fig. 5 shows the effect of increasing chromium additions in producing resistance to oxidation by air at increased temperatures. The maximum service temperature for any steel will depend entirely on the oxidation rate which may be tolerated in service. For pressure parts, such as superheater elements in contact with steam, a sufficiently safe standard for construction may be based upon an oxidation rate of approximately 0.005 in. penetration per year, while for heavy support parts higher reaction rates may be tolerated.

The effect of silicon in improving resistance of iron alloys to scaling in straight air atmospheres is indicated in Fig. 6 (100 hr test).

Fig. 7 presents scaling data on a number of commercial alloy steels for the temperature range of 950 to 1200 F, representing normal metal temperatures on the steam side of high temperature superheater elements. These data are based upon oxidation by air but with suitable interpretation may serve as a basis for selection of an alloy steel for superheater service. The effect of silicon in improving resistance of chromium bearing steel to scaling by air should again be noted.

Definite laboratory data on the oxidation of metals by high temperature steam are meager. Some investigations of the scaling of steel by steam at very high temperatures have been made at very high temperatures, about 1700 to 1900 F. The rate of attack at these temperatures is considerably greater for steam than for air. This has been considered as due to a more porous oxide formed by steam. The porosity has been suggested as being due to the presence of some hydrogen, produced by the reduction of H_2O by iron, so that alternate oxidation and reduction occurs. Recent tests at The Detroit Edison Co. of some 18 steels under 1100 F steam conditions have confirmed the superior oxidation resistance of low chromium steels (below 12 per cent chromium) for superheater service at this temperature. The role of silicon in providing increased resistance to steam attack has not been substantiated, but tests on such steels are now being conducted.

Mention should be made of the very extensive steam oxidation tests now being conducted at Purdue University for the ASME special research committee on critical pressure steam boilers.

Mention has been made of the decrease in strength of plain carbon steel with temperatures in excess of 650 F, and of the superiority of carbon-molybdenum steel within the range of 650 F to 950 to 1000 F by reason of its enhanced resistance to creep. Creep may be defined as the slow deformation of a metal when subjected for long periods of time to conditions of high temperature and stress. Creep tests are usually based on time-extension

Table 1—Methods by Which Alloying Elements Increase Strength of Steel (Elements Arranged According to Relative Intensity of Their Effects)

A. Solid Solution Hardness:	C. Carbide Nature and Form:
1. Phosphorus	1. Vanadium
2. Silicon	2. Chromium
3. Manganese	3. Molybdenum
4. Nickel	4. Manganese
5. Chromium in excess of that required to form carbides	D. Hardenability on Cooling From Above Critical Range:
6. Molybdenum in excess of that required to form carbides	1. Carbon
7. Vanadium in excess of that required to form carbides	2. Phosphorus
8. Copper	3. Manganese
B. Refinement of Grain:	4. Chromium
1. Vanadium	5. Molybdenum
2. Molybdenum	6. Vanadium
	7. Nickel
	8. Silicon
	E. Precipitation Hardening:
	1. Copper

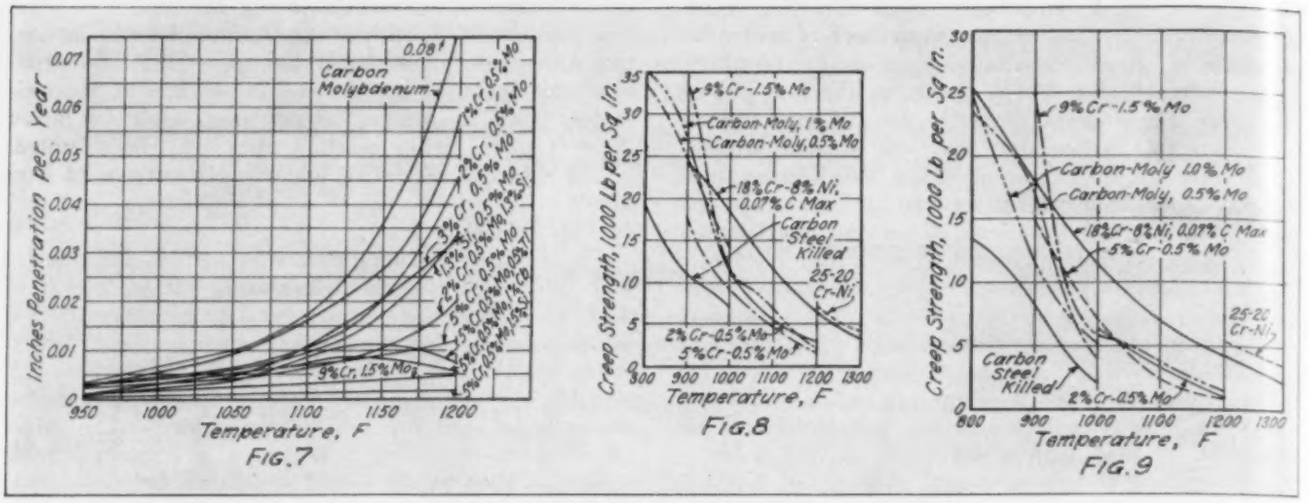


FIG. 7 FIG. 8 FIG. 9

Fig. 7—Relative oxidation resistance (in air) of some commercial superheater steels . . . . . Fig. 8—Curves showing comparative creep strength of alloy steels—rate 1 per cent in 10,000 hr . . . . . Fig. 9—Curves showing comparative creep strength of alloy steels —rate 1 per cent in 100,000 hr

observations over long periods on test pieces subjected to constant load and temperature. From time-elongation curves a linear relationship between log of stress and log of rate of creep may be plotted, from which the stress giving any definite rate of creep, such as 1 per cent in 10,000 hr or in 100,000 hr may be obtained. Such creep values (Figs. 8 and 9) serve as a basis of design for high temperature, high pressure equipment.

More recently rupture-stress-time tests have been conducted, which again show a linear logarithmic relationship. These tests have indicated that many steels at temperatures and time periods in excess of a certain amount fail in an intergranular manner with reduced ductility. While no long time tests, extending to many thousands of hours, have been completed, it is assumed or predicted that with increasing time to fracture decreased ductility is obtained. It has even been suggested that if the tests were conducted at stresses sufficiently low to cause rupture in 10,000 or 100,000 hr, the total elongation might approach or be less than 1 per cent, which is equivalent to the total creep presented for creep values. However, the stresses causing rupture with accompanying intergranular fracture and impaired ductility are much higher than the corresponding creep values of the steel, and the intergranular fracturing of a steel under excessive loads should not condemn it for use under much lower stresses than that causing rupture in this type of testing.

Internal microstructural changes of the steel may occur as a result of exposure to the service temperature for long periods of time or to accidental overheating or to rates of cooling through a temperature range in the normal operation of a high temperature equipment. Such changes may result in an impairment of the physical properties of the steel.

Excessive heating of the pearlitic steels to within the region of the lower critical point may result in spheroidization of the carbides with consequent reduction in the creep resistance of the steel at normal temperature operation. For example, carbon-molybdenum steel heated at 1200 F for two weeks, showed a creep value of less than 50 per cent of the creep value of annealed material.

Temper brittleness may develop after long time heating at approximately 900 F, as in the case of the high chromium irons of above approximately 15 per cent or as a result of slow cooling from this temperature range, as in the case of straight 5 per cent chromium steels (without molybdenum). Such brittleness results from a structural change involving the precipitation of a hard intermetallic compound, as for example the precipitation of a network of carbides at the grain boundaries.

The coefficient of thermal expansion of alloys is of importance in the design and operation of high pressure, high temperature equipment. Fatigue failure may develop as a result of repeated stresses induced by varying and repeated thermal gradients established in the structure, as for example in the feed water connection to a heavy drum or around inlet nozzles for cold dousing operations in oil refinery cracking stills.

In many structures it is necessary to join two steels which may have widely different coefficients of thermal expansion, as for example pearlitic or ferritic steels, such as plain carbon steels and austenitic nickel-chromium steels such as 18 per cent chromium, 8 per cent nickel. The latter alloy steel possesses a coefficient of expansion 50 per cent greater than that of plain carbon steel, and on heating a composite structure where the two steels are intimately joined, as by welding, high internal stresses are produced with possibility of failure of some type.

The weldability of the various special steels for high temperature, high pressure service represents the most important fabrication problem. Table 1 presents the methods by which the individual elements confer increased strength to steel. Consideration of the thermal conditions under which fusion welds are made will immediately indicate that the factor of hardenability on cooling from welding temperature is the most important property of the steel to be considered with respect to the alloying elements present (item *D*). Carbon exerts the most pronounced hardening effect, so that in all of the pearlitic alloy steels its content should be limited, generally to a maximum of 0.15 per cent carbon. This limit, however, is not definite and will vary from this value in both directions, dependent upon the amount of additive alloying elements present, particularly those which are quite effective in conferring hardenability, such as manganese and chromium.

The Psychrometric Chart

Its Application and Theory

By **William Goodman***

IN order to draw a ratio line on the psychrometric chart, only the initial condition of the air and the value of the moisture ratio need to be known. Knowing the moisture ratio, the value of the ratio angle θ† of Fig. 3 can be found in Table 1. With both the initial point *1* and the ratio angle θ known, the ratio line *1-3* can be drawn on the psychrometric chart by means of a protractor. Once the line has been drawn, all of the possible final states of the air *leaving* the chamber can be read along the ratio line. The following example illustrates the method of using these ratio lines.

Example 1: The heat flowing into a chamber amounts to 3500 Btu per min. In addition, 3 lb per min of steam is added to the air. The steam is saturated at an initial pressure of 5 lb ga. The quantity of dry air flowing through the chamber amounts to 413 lb per min. The initial state of the air entering the conditioning chamber is 74 deg dry bulb and 59 deg dew point. The barometric pressure is 29.92 in. Find the final dry bulb and dew point temperatures of the air leaving the chamber.

Solution: From the Keenan and Keyes steam tables,[2] for saturated steam at 5 lb ga, $h_0 = 1156$ Btu per lb.

Using equation 5,

$$q = \frac{3500}{3} + 1156 = 2323 \text{ Btu per lb of moisture added to the air}$$

Referring to Table 1, for $q = 2323$ Btu, $\theta = 48°\text{–}34'$. Locate point *1* in Fig. 4 at the initial state of 74 deg dry bulb and 59 deg dew point temperature. Draw a horizontal line through point *1* as shown in Fig. 4. From this horizontal line, measure the angle $\theta = 48°\text{–}34'$ downward in a counterclockwise direction; then draw line *1–3*.

From Chart A, at point *1*,

$$w_1 = 0.01066 \text{ lb per lb of dry air}$$

From equation 2,

$$w_2 = w_1 + \frac{W_0}{G}$$

$$= 0.01066 + \frac{3}{413}$$

$$= 0.01792 \text{ lb per lb of dry air}$$

Draw a horizontal line on the chart for $w_2 = 0.01792$ lb. This line intersects the sloping line *1–3* at point *2* (Fig. 4). Hence, point *2* represents the final condition of the air leaving the chamber. The final dry bulb temperature of the air as read at point *2* of Fig. 4 is 110 deg and the dew point temperature is 73.7 deg.

If the quantities of heat and moisture added to the air are constant, the final condition of the air is represented by a point that lies on the ratio line *1-3* of Fig. 3. The final enthalpy of the air leaving the chamber will be different for each point on the ratio line. In equation 1 the right-hand member is constant because the quantities of heat and moisture added to the air are assumed to be constant. Consequently, for any given initial enthalpy, as the final enthalpy of the air changes along the ratio line, the weights of air, G, that should flow through the chamber must change. From equation 1, the greater the value of the quantity $(h_2 - h_1)$, the smaller must be the weight of air flowing through the chamber because the right-hand member of this equation is constant. Therefore, for each final point selected on the ratio line, a different weight of air must be provided in order to absorb the heat and moisture gains of the conditioned space. This has been discussed fully in an earlier article.[1]

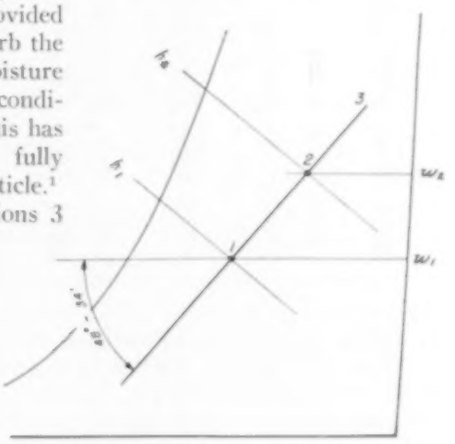


Fig. 4—Illustration used in solving example 1

So far, equations 3 and 5 have been discussed as if they were valid only when moisture is added to the air in the form of steam. Manifestly, these equations would still be true even if moisture in liquid form, or a mixture of liquid and steam, were sprayed into the air. In such a case, the value of h_0 would simply correspond to the enthalpy of the water or of the mixture of water and steam being sprayed into the air.

Example 2: Take the conditions of example 1 except that, instead of steam, water at a temperature of 90 deg is sprayed

In addition to the elementary uses of the psychrometric chart—such as showing the relationships between dry bulb, wet bulb, and dew point temperatures, and relative humidity—it provides a simple and enlightening means of analyzing many complex problems in comfort and process air conditioning, including drying. . . . New psychrometric charts covering the low, middle and high temperature ranges, with saturation curves for various barometric pressures, are presented here. The general theory of the psychrometric chart is discussed in detail, and the methods of solving problems by the use of charts are described. . . . These new charts facilitate solution of most problems and lead to clearer understanding of the fundamentals

*The Trane Co. Member of Board of Consulting and Contributing Editors.

[2]The Keenan and Keyes steam tables, published by John Wiley & Sons, Inc., New York, are used in the solution of all examples.

†See table of symbols, p. 358, June issue.

Part 2. Part 1 was published in HEATING, PIPING AND AIR CONDITIONING, June, 1939, pp. 357-360.

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Table 1—Ratio Angles for Various Values of Moisture Ratio

Moisture Ratio q	Ratio Angle θ	Moisture Ratio q	Ratio Angle θ	Moisture Ratio q	Ratio Angle θ	Moisture Ratio q	Ratio Angle θ	Moisture Ratio q	Ratio Angle θ	Moisture Ratio q	Ratio Angle θ
Inf.	0	1780	63° —18′	940	—85° —10′	490	—68° —14′	40	—54° —28′	—1000	—34° —45′
50,000	1° —40′	1760	63 —57	930	—84 —46	480	—67 —53	30	—54 —12	—1100	—33 —29
40,000	2 — 6	1740	64 —36	920	—84 —22	470	—67 —32	20	—53 —56	—1200	—32 —18
30,000	2 —50	1720	65 —15	910	—83 —58	460	—67 —12	10	—53 —41	—1300	—31 —12
20,000	4 —19	1700	65 —55	900	—83 —34	450	—66 —52	0	—53 —25	—1400	—30 — 9
18,000	4° —50′	1680	66° —35′	890	—83° —11′	440	—66° —31′	0	—53° —25′	—1500	—29° —10′
16,000	5 —28	1660	67 —16	880	—82 —47	430	—66 —11	— 10	—53 —10	—1600	—28 —15
14,000	6 —18	1640	67 —57	870	—82 —23	420	—65 —51	— 20	—52 —54	—1700	—27 —22
12,000	7 —26	1620	68 —39	860	—82 — 0	410	—65 —31	— 30	—52 —39	—1800	—26 —33
10,000	9 — 5	1600	69 —20	850	—81 —36	400	—65 —14	— 40	—52 —24	—1900	—25 —46
9,000	10° —12′	1580	70° — 3′	840	—81° —13′	390	—64° —51′	— 50	—52° — 9′	—2000	—25° — 2′
8,000	11 —38	1560	70 —45	830	—80 —49	380	—64 —32	— 60	—51 —54	—2200	—23 —40
7,000	13 —32	1540	71 —28	820	—80 —26	370	—64 —12	— 70	—51 —39	—2400	—22 —27
6,000	16 — 9	1520	72 —12	810	—80 — 3	360	—63 —53	— 80	—51 —24	—2600	—21 —20
5,000	19 —57	1500	72 —56	800	—79 —39	350	—63 —33	— 90	—51 —10	—2800	—20 —19
4800	20° —55′	1480	73° —40′	790	—79° —16′	340	—63° —14′	—100	—50° —55′	—3000	—19° —24′
4600	22 — 0	1460	74 —24	780	—78 —53	330	—62 —55	—110	—50 —41	—3200	—18 —33
4400	23 —11	1440	75 — 9	770	—78 —30	320	—62 —36	—120	—50 —26	—3400	—17 —46
4200	24 —29	1420	75 —54	760	—78 — 7	310	—62 —17	—130	—50 —12	—3600	—17 — 3
4000	25 —56	1400	76 —40	750	—77 —44	300	—61 —58	—140	—49 —58	—3800	—16 —23
3800	27° —34′	1380	77° —25′	740	—77° —21′	290	—61° —40′	—150	—49° —44′	—4000	—15° —46′
3600	29 —23	1360	78 —11	730	—76 —58	280	—61 —21	—160	—49 —30	—4200	—15 —12
3400	31 —26	1340	78 —57	720	—76 —35	270	—61 — 3	—170	—49 —16	—4400	—14 —40
3200	33 —46	1320	79 —44	710	—76 —12	260	—60 —44	—180	—49 — 2	—4600	—14 —10
3000	36 —24	1300	80 —31	700	—75 —50	250	—60 —26	—190	—48 —49	—4800	—13 —42
2900	37° —52′	1280	81° —17′	690	—75° —27′	240	—60° — 8′	—200	—48° —35′	—5000	—13° —16′
2800	39 —25	1260	82 — 4	680	—75 — 5	230	—59 —50	—220	—48 — 8	—6000	—11 —27
2700	41 — 6	1240	82 —52	670	—74 —42	220	—59 —32	—240	—47 —42	—7000	—10 — 3
2600	42 —53	1220	83 —39	660	—74 —20	210	—59 —14	—260	—47 —16	—8000	— 8 —58
2500	44 —49	1200	84 —27	650	—73 —58	200	—58 —56	—280	—46 —50	—9000	— 8 — 5
2400	46° —52′	1180	85° —15′	640	—73° —35′	190	—58° —39′	—300	—46° —24′	—10,000	— 7° —22′
2300	49 — 5	1160	86 — 2	630	—73 —13	180	—58 —21	—320	—45 —59	—11,000	— 6 —46
2200	51 —27	1140	86 —50	620	—72 —51	170	—58 — 4	—340	—45 —35	—12,000	— 6 —15
2100	53 —59	1120	87 —38	610	—72 —29	160	—57 —47	—360	—45 —10	—13,000	— 5 —48
2000	56 —42	1100	88 —26	600	—72 — 8	150	—57 —30	—380	—44 —46	—14,000	— 5 —25
1980	57° —16′	1080	89° —14′	590	—71° —46′	140	—57° —13′	—400	—44° —23′	—15,000	— 5° — 5′
1960	57 —50	1061	90 — 0	580	—71 —24	130	—56 —56	—420	—43 —59	—20,000	— 3 —53
1940	58 —25	1040	—89 —10	570	—71 — 3	120	—56 —39	—440	—43 —36	—30,000	— 2 —38
1920	59 — 0	1020	—88 —21	560	—70 —41	110	—56 —22	—460	—43 —13	—40,000	— 2 — 0
1900	59 —35	1000	—87 —33	550	—70 —20	100	—56 — 5	—480	—42 —51	—Inf.	0
1880	60° —11′	990	—87° — 9′	540	—69° —59′	90	—55° —49′	—500	—42° —29′		
1860	60 —48	980	—86 —45	530	—69 —37	80	—55 —32	—600	—40 —43		
1840	61 —25	970	—86 —22	520	—69 —16	70	—55 —16	—700	—39 — 4		
1820	62 — 2	960	—85 —58	510	—68 —55	60	—55 — 0	—800	—37 —32		
1800	62 —40	950	—85 —34	500	—68 —34	50	—54 —44	—900	—36 — 5		

into the air stream. All of this water is evaporated into the air. Find the final dry bulb and dew point temperatures of the air leaving the chamber.

Solution: For water at 90 deg, $h_o = 58$ Btu.

Using equation 5,

$$q = \frac{3500}{3} + 58$$

$= 1225$ Btu per lb of moisture added to the air

Referring to Table 1 for $q = 1225$, $\theta = 83°-27'$. Using Chart A, locate the initial condition of 74 deg dry bulb and 59 deg dew point temperature and mark it point *1* as was done in Fig. 4. Draw a horizontal line through point *1*, and from this horizontal line measure the angle $\theta = 83°-27'$ downward in a counterclockwise direction; then draw a line *1–3* as was done in Fig. 4. From example 1, $w_2 = 0.01792$ lb of moisture. Draw a fiorizontal line on the chart for $w_2 = 0.01792$ lb. This line will intersect the ratio line *1–3* at point *2*. The final dry bulb temperature of the air as read at point *2* of a chart similar to Fig. 4 is 77.8 deg, and the dew point temperature is 73.7 deg.

The moisture ratio line is also useful for determining the amount of heat required if the initial and final conditions of the air are known. If points *1* and *2* of Fig. 3 are known and a straight line is drawn between them, the angle θ of Fig. 3 can be measured by means of the protractor. Then referring to Table 1 for the measured value of the ratio angle, the moisture ratio can be found. As is evident from equations 4 and 5, the numerical value of the moisture ratio is, of course, the quantity of heat added per pound of moisture evaporated. In general, the amount of heat added or removed from the air can be found more simply by taking the difference between the enthalpy of the air in the final and initial conditions. There are, however, certain types of problems that cannot be solved in any other way except by first finding the moisture ratio. The following example illustrates the solution of such a problem.

Example 3: Air at a barometric pressure of 24 in. enters the conditioning chamber at a dry bulb temperature of 58 deg and a dew point temperature of 54 deg. The chamber is to be maintained at a dry bulb temperature of 85 deg and a dew point temperature of 69 deg. The heat gain of the chamber is 11,300 Btu per min. Saturated steam at a pressure of 5 lb ga and a quality of 80 per cent is to be sprayed into the chamber for humidifying purposes. Find the quantity of air and steam that must be supplied to the chamber in order to maintain the required conditions.

Solution: The condition of the air leaving the room will be the same as the condition to be maintained inside the room. Inasmuch as neither the air nor steam quantities are known, equations 1 and 2 cannot be used to solve this problem. Equation 5 must be used to find the quantity of steam required after first measuring the value of the moisture ratio on the chart or computing its value by means of equation 4.

Locate points *1* and *2* on the psychrometric chart as illustrated in Fig. 3, and draw the ratio line between the two points. Then by measurement the ratio angle is found to be 58°–51′, and by using Table 1 the moisture ratio, q, is found to be 1925 Btu per lb of moisture added to the air. Inasmuch as the steam supplied to the chamber has a quality of 80 per cent—that is, each pound of steam sprayed into the air consists of 0.8 lb of saturated steam and 0.2 lb of water—the enthalpy of the steam is figured as follows:

$$h_o = 0.2 \times 196.2 + 0.8 \times 960.1$$

$= 807.3$ Btu per lb, enthalpy of steam sprayed into the air

Using equation 5,

$$1925 = \frac{11{,}300}{W_o} + 807.3$$

Therefore,

$$W_o = \frac{11{,}300}{1925 - 807.3} = 10.11 \text{ lb per min of moisture required}$$

Either equation 1 or 2 can now be used to find the weight of air that must be supplied to the room. The values of w_1 and

w_2 can be read from the psychrometric chart at points *1* and *2*. Using equation 2,

$$G(0.0191 - 0.01109) = 10.11$$

$$G = \frac{10.11}{(0.0191 - 0.01109)} = 1262 \text{ lb per min of air required}$$

Notice in Table 1 that many of the values of the ratio angle θ are negative. Such negative values mean that the ratio angle is to be measured upward from the horizontal line in a clockwise direction as shown in Fig. 5. This is done for convenience in order to avoid awkward manipulation of the protractor when drawing the ratio lines. Notice that although negative values of the moisture ratio are listed in Table 1, this does not necessarily mean that the corresponding values of the ratio angle will also be negative. Negative values of the ratio angle will occur with either positive or negative values of the moisture ratio, q.

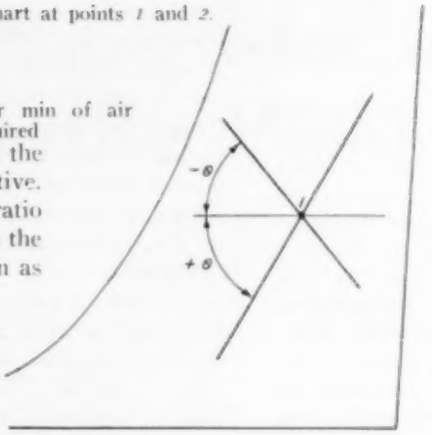


Fig. 5—Measurement of positive and negative ratio angles

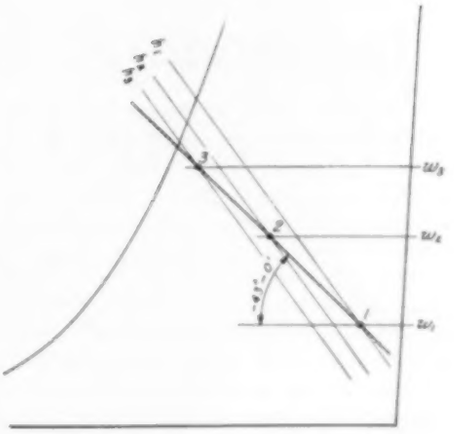


Fig. 6—Illustration used in solving example 4

For the simple case illustrated in Fig. *2*, where both heat and moisture are added to the air flowing through the chamber, there is no possibility of confusion in computing the moisture ratio. On the other hand, when heat is added while moisture is removed, or vice versa, there is a possibility of incorrectly computing the moisture ratio unless the method of handling such cases is fully understood.

When either heat or moisture flow into a chamber, and are eventually added to the air flowing through the chamber, the numerical value for the heat and moisture quantities should be preceded by a *plus* sign. On the other hand, when either heat or moisture are lost from a chamber and, hence, in turn surrendered by the air flowing through the chamber, the numerical value of the heat quantity, Q, and the moisture quantity, W_o, must be preceded by a *negative* sign. When heat is lost while moisture is gained, a minus sign is used in front of Q, and a plus sign in front of W_o. If the situation is reversed and heat is gained while moisture is lost, a plus sign should be used in front of Q, and a minus sign in front of W_o. In all cases, a *plus* sign is always used in front of the numerical value of h_o, the enthalpy of the moisture entering or leaving the chamber. The following example illustrates one of the cases just discussed.

Example 4: In a certain drying process the quantity of heat lost from the chamber is 1500 Btu per min. The product entering the chamber carries in 3 lb per min of water at a temperature of 60 deg, and it is to leave the chamber bone dry. The dry bulb temperature to be maintained in the chamber is 85 deg, and the dew point temperature is 52 deg. Find the required initial dry bulb and dew point temperatures of the air supply if the total quantity of dry air passing through the chamber is 1200 lb per min. The barometric presure is 29.92 in.

Solution: Inasmuch as heat is lost, $Q = -1500$ Btu per min. Inasmuch as moisture is added to the air, $W_o = +3$ lb per min. For water at 60 deg, $h_o = 28$ Btu per lb.

By means of equation 5,

$$q = \frac{-1500}{+3} + 28 = -472 \text{ Btu, heat } \textit{loss} \text{ per lb of moisture } \textit{added}$$

In Fig. 6 let point *2* represent the room conditions of 85 deg dry bulb and 52 deg dew point. Referring to Table 1, for $q = -472$, $\theta = -43°\text{-}0'$. Draw the ratio line *1–2–3* through point *2*.

Referring to Fig. 6,

$$w_2 = 0.00823 \text{ lb per lb of dry air}$$

Using equation 2,

$$w_1 = w_2 - \frac{W_o}{G}$$

$$w_1 = 0.00823 - \frac{3}{1200} = 0.00573 \text{ lb per lb of dry air}$$

Locate point *1* on the ratio line *1–2–3* of Fig. 6 where the horizontal moisture line of 0.00573 lb intersects the ratio line. The required initial condition of the air as read at point *1* is 101.2 deg dry bulb and 42.5 deg dew point temperature.

In the preceding example, the air must replace the heat lost from the chamber. In addition the air must carry away the moisture evaporated from the product and at the same time provide the necessary heat to vaporize the water. If the situation in the preceding example had been reversed—that is, heat was added to the chamber and moisture lost from the chamber—the answer obtained for the moisture ratio, q, and for the ratio angle θ would be exactly the same. However, the required initial dry and wet bulb temperatures of the air supply would lie, say, at point *3* on the ratio line *1–3*, because in this case the dry bulb temperature of the air must rise as it absorbs the heat gain, and its dew point temperature must fall as it surrenders moisture to replace the moisture lost.

Example 5: Take the conditions of example 4 except that the heat gain of the chamber is 1500 Btu per min and there is a loss of moisture of 3 lb per min.

Solution: By means of equation 5,

$$q = \frac{1500}{-3} + 28 = -472 \text{ Btu heat } \textit{gain} \text{ per lb of moisture } \textit{lost}$$

(The sign in front of $h_o = 28$ is always *plus* as previously stated in the text.)

Therefore, $\theta = -43°\text{-}0'$ as in example 4. With point *2* and the angle θ both known, the line *2–3* can be drawn.

Using equation 2,

$$w_1 = w_2 - \frac{W_o}{G}$$

$$= 0.00823 - \frac{-3}{1200}$$

$$= 0.00823 - (-0.0025)$$

$$= 0.01073 \text{ lb}$$

The required initial condition of the air is read at point *3*

where the line of absolute humidity of 0.01073 lb intersects the ratio line *1–3*. The required initial dry bulb temperature of the air supply as read at point *3* is 69.0 deg, and the required initial dew point temperature is 59.2 deg.

A negative value of the moisture ratio simply means that heat is removed while moisture is added, or vice versa. As can be seen from the preceding two examples, regardless of whether heat is added while moisture is removed, or the reverse is true, as long as the numerical values for the heat and moisture quantities are the same in both cases, the values of the moisture ratio and the ratio angle will also be the same. Even without any computations, the common sense of the situation will usually indicate where the point representing the required initial state of the air must lie in relation to the point representing the state to be maintained in the conditioned space.

Any process in which energy is added to or removed from a substance can be analyzed by the method used to analyze the simple case of Fig. 2. However, the right-hand member of the equation will not always reduce to a constant value. In such a case, even though the equation is similar to the one for Fig. 2, all of the points representing the possible final states of the air will *not* fall on a straight line on the psychrometric chart. It is important to bear in mind the fact that only if the right-hand members of equations 3 and 5 are constant will the points representing the final states of the air fall on a straight line on the psychrometric chart. This is discussed in more detail later.

The use of ratio lines for solving different types of problems that arise in practical air conditioning work was described in an earlier article[1] In this earlier article the slope of the ratio line was found by first computing the *sensible heat ratio*, which is discussed in the next section. The ratio angle and the location of the ratio line on the chart can be determined either from the sensible heat ratio or from the moisture ratio.

[*To be continued*]

Installation and Servicing of Air Conditioning Jobs

IN A TALK on installation and servicing of air conditioning jobs of less than 50 tons capacity at the recent air conditioning conference at the University of Texas, A. F. Avera, Air-Con Engrg. Corp., stressed the need of an itemized estimate sheet. Cost estimates should be broken down into a sufficient number of items to permit the bookkeeper to charge up job costs intelligently and to determine after the job is completed which of the items were incorrectly estimated, he said. An estimate which a cost accountant can fathom should include the following:

Name of customer. Date. Compressor—speed, motor horsepower and current characteristics. Air conditioning unit—row coil, fan speed, horsepower, current characteristics. Filter section. Expansion valves. Fan starter. Room thermostat and other controls. Compressor foundation. Unit hangers or supports. Cooling tower—cfm, size, etc. Cooling tower foundations. Condenser water pump—size, capacity, horsepower, current characteristics. Pump starter. Refrigerant piping. Cork covering. Duct work. Duct insulation. Grilles and outlets. Painting ducts, if any. Cutting and patching. "Freon" charge. Labor and expenses. Rigging. Water and drain lines. Electric wiring. Freight and drayage. (The labor and expense item can be broken down on a separate sheet and entered on the estimate as one item.)

Mr. Avera concluded with the following pointers:

Refrigerant piping should be sized to carry the tonnage at the right velocity; oil travels through the system in solution with "Freon," and it is necessary to have lines properly sized and trapped to bring the oil back to the compressor.

The improper adjustment of expansion valves will affect oil circulation. Therefore, much care should be taken in the adjustment of valves.

The prime essential to the installation of a job after the piping has been tested is to thoroughly dehydrate the system. This must be done with a vacuum pump and takes time. It will pay big dividends, however, in low compressor repair bills.

V-belt drives should be carefully lined up. Compressors and fans will run quieter and the belts will last longer if the drive is in line.

Starters should have proper heater coils; it is often necessary when a compressor is installed in a hot place to provide heater coils with 30 per cent higher amperage rating than the motor.

Structural supports and foundations should be given careful consideration. Higher compressor and fan speeds make proper isolation measures to reduce noise and vibration important.

Evaporative condensers and cooling towers should be carefully located.

An air conditioning job certainly cannot work when the filters are so completely stopped up that the fan is not delivering the required air.

Rating of Heat Transmitters

In Samuel R. Lewis' article in the May HEATING, PIPING AND AIR CONDITIONING (a reader writes), appears the following paragraph: "My hope for a rating of heat transmitters which warm the air of rooms, based on the heat delivered by the transmitter rather than on the heat input to it, has not been realized but there are indications that it will arrive eventually."

As this subject has been attacked from many angles and is of great interest to me, I would appreciate very much Mr. Lewis' viewpoint and the method he feels would produce the proper rating of such transmitters.

Reply—I proposed seriously, 10 years ago, that certain accredited laboratories in various accessible locations, preferably at state universities, be provided with standard test rooms or booths where ratings of various heat transmitters in producing comfort within the rooms

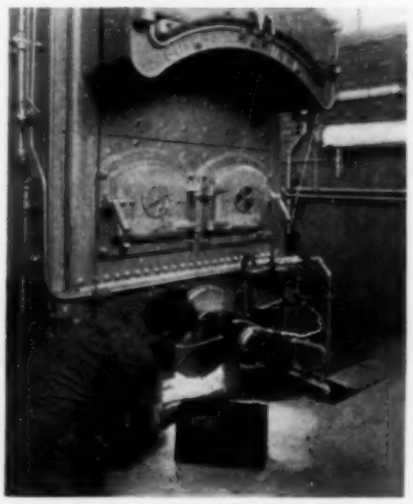
[*Concluded on p.* 431]

Regular Supervision of Apartment Hotel Heating Plants Cuts Costs

By Kalman Steiner*

THE fuel consumption for heating buildings is always a topic of prime importance to heating engineers. There is discussed here some material secured in a rather unusual manner—or at least arising from an opportunity which does not occur any too frequently in practice. It consists of data on the fuel consumption of a group of large residential buildings in Chicago which are interesting because of the opportunity to observe the effect upon fuel consumption of a closely checked supervisory service that was instituted in an effort to reduce fuel consumption.

This group of buildings has been managed by the same realty firm for a number of years, so that complete records are available of the fuel purchases. For this and a number of other reasons, the buildings constituted an ideal group for this sort of experiment. The objective was to utilize the existing heating apparatus more effectively, and to supplement the efforts of the men in the buildings who had charge of the operation of the heating systems. At the beginning of the 1937-38 heating season the management entered into a supervisory agreement, under which the contracting firm was to make a monthly inspection and check up of each boiler plant, and submit a report.



A monthly call was made at each building by an experienced service engineer

All the buildings use oil fuel. All were erected during the late building boom of the '20's. Six are corridor type apartment hotels, one is a residential hotel, and one is a store and apartment building. An analysis of the buildings appears in Table 1, and points of similarity and difference are enumerated.

Monthly Call by Service Engineer

The monthly call at each building was made by an experienced service engineer versed in oil burner construction, combustion, and heating plants. He first thoroughly examined the oil burner mechanism, cleaning the strainers, nozzles, fan, etc.; then he checked the burner adjustments to see that everything was working smoothly. He then tested the combustion gases for CO_2, stack temperature, and draft. As a result of these tests the draft conditions were altered if necessary to meet the combustion requirements. The final work was to test all safety controls: low water cutout, stack or combustion safety, and pressurestat. There was little of importance about the oil burner that was likely to have escaped attention after this routine, but the service engineer made it a point to discuss the installation with the janitor or engineer in each case so that any irregularities not apparent from the inspection could be noted and the required action taken.

After this call was completed, a full report was turned in to the office of the contracting firm, where a composite monthly report was prepared for the management company. This composite report discussed the trend of the fuel consumption and pointed out any conditions necessitating repair. All repairs were attended to promptly by the managing company so that all features of the apparatus were always maintained in good working order. Any suggestions contained in the reports for changes in operating routines or practices were carried through, the respective operators and building managers giving their complete coöperation.

Analysis at End of First Season

At the end of the trial heating season a complete analysis and a report was drawn up to show the tangible results of the monthly check up service. The fuel consumption data, and the deductions made from them, are given below.

Since all of the buildings use fuel oil for hot water heating as well as heating the building, the problem of the basis to use for computation and calculation was presented. It was decided to eliminate from the comparative analysis the data for the months from May to September, since during that period the fuel consumption is largely for hot water heating and so is not subject to comparison from a degree day standpoint. Nor was any attempt made to break down the fuel consumption from October through April into respective portions for heating and hot water.

Table 2 gives fuel consumption and gallons used per degree day for the eight buildings. These figures are for October through April.

A closely checked supervisory service was instituted for a group of eight buildings in an effort to reduce fuel consumption; the group comprised six corridor type apartment hotels, a residential hotel, and a store and apartment building. . . . The results obtained indicate worthwhile possibilities in cutting heating costs by expert and specialized supervision

*Ace Engineering Co.

Degree days for the three periods involved were: 1935-36, 6,644; 1936-37, 5,956; 1937-38, 5,618. It will be observed that the first season of the three was the coldest, the second season intermediate, and the third the mildest. As can be expected, the two less severe winters show lower fuel consumption. But it is necessary to examine the variation of consumption with degree days more closely in order to determine what relationship exists between the variables.

Reference to the charts shows the existence of a pronounced general form of the curves resulting from plotting gallons per degree day against total degree days per month. Clearly, the fuel consumed *per degree day* diminished as the total degree days per month increased. In other words, heat can be supplied more economically as the demand for heat increases. Putting it still a third way, the unit cost of production decreases as the total quantity of production increases. It was necessary to plot the data in this manner in order to bring out this relationship.

Referring to Table 2, in five cases the gallons per degree day increased from the first to the second year, in two cases it decreased, and in the remaining building it showed no change. It is possible, of course, that some of the deviation may arise because of errors in the reported gallonage figures or to carrying of fuel stocks from one period into the next. At least, so far as was known, there were no changes made in the heating equipment or routine that should have caused the deviation.

Table 1—Data on the Eight Buildings

Building No.	1	2	3	4	5	6	7	8
Age of building, years	10	11	14	11	11	10	12	15
Type of building	A	A	B	A	A	C	A	A
Number of floors	22	7½	3	7	13	5½	6	6½
Number of apartments	64	87	26	85	49	2	37	66
Total apartment rooms	325	190	56	145	267	4	64	134
Number of hotel rooms						80		
Number of bathrooms	126	87	26	85	120	42	37	66
Number of stores			5					
Extra hot water load			D			E		
Number and kind of boilers	2-F	1-G	2-F	1-F	2-F	1-F	1-F	1-F
Capacity of boilers, 100 sq ft	141	110	93	110	67	73	57	73
Kind of oil burner	H	H	I	H	H	H	H	H
Capacity of oil burner, 100 sq ft	180	280		80	120	120	60	80
Kind of hot water system	J	K	L	K	L	K	K	K
Capacity of hot water tank, 100 gal	20	5	6	20		15	15	15
Type of heating system	M	M	N	M	M	M	N	M
Estimated radiation, 100 sq ft	178	89	30	71	150	32	28	55
Estimated building volume, 1000 cu ft	1324	655	250	450	912	322	205	385
Control of burner	O	O	P	O	O	O	O	O
Control of heating system	Q	R	S	Q	R	Q	Q	R
Control of draft	T	U		U	V	T	T	U
Control of hot water	W	X	Y	X	X	X	X	X

KEY

A—Corridor type fireproof apartment.
B—Apartment and store building.
C—Fireproof hotel, women guests only.
D—Drug store serving lunches.
E—Tea room.
F—Welded steel tubular.
G—Brick set steel firebox.
H—Horizontal rotary.
I—Vertical rotary.
J—Direct steam coils in tank.
K—Heater submerged in boiler water.
L—Indirect heater using boiler water.
M—Vacuum heating.
N—One pipe gravity.
O—Automatic ignition.
P—Manual ignition.
Q—Combination outdoor bulb with condensate return bulb.
R—Combination outdoor bulb with radiator bulb.
S—Constant steam pressure on boiler.
T—Hydraulic overfire regulator.
U—Balanced barometric controller.
V—Electric overfire regulator.
W—Thermostatic steam valve.
X—Aquastat in summer.
Y—No control in winter; use gas in summer.

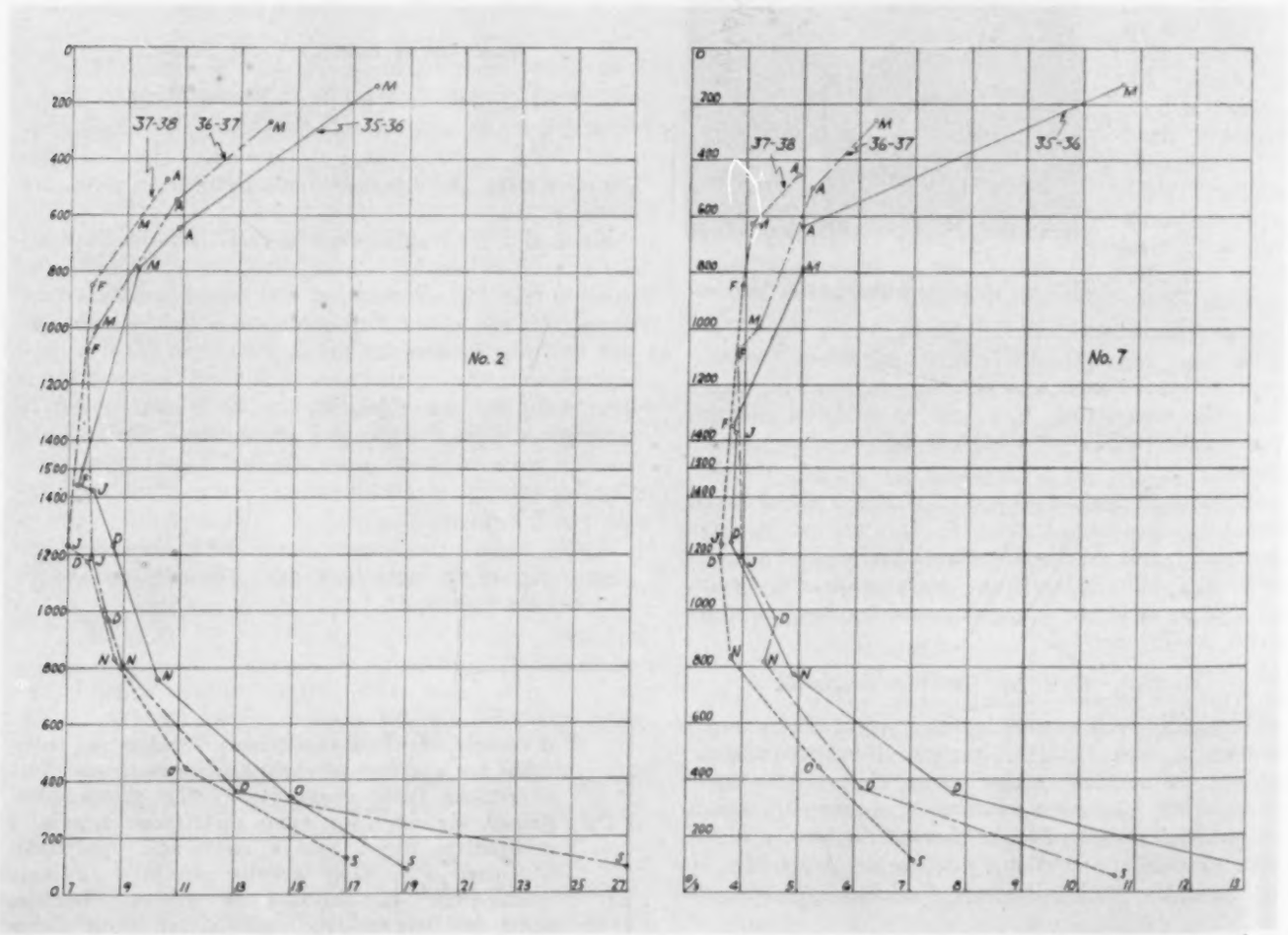


Table 2—**Fuel Consumption and Gallons Used per Degree Day**

Building	1935–36		1936–37		1937–38	
	Gal	Gal/Dd	Gal	Gal/Dd	Gal	Gal/Dd
1	111,997	16 90	112,591	18 90	98,113	17 50
2	59,887	9 03	51,822	8 70	47,326	8 42
3	31,102	4 69	26,097	4 38	26,640	4 75
4	48,300	7 28	44,994	7 55	44,341	7 86
5	83,233	12 55	74,928	12 55	71,095	12 65
6	34,754	5 24	34,121	5 72	30,673	5 46
7	28,884	4 36	26,320	4 41	22,338	3 98
8	39,855	6 02	37,858	6 35	35,555	6 32
Totals	438,012	66 11	408,731	68 50	376,081	67 00

But a sharp change occurs in comparing the third year with the second. Again there has been a drop in degree days, so that it would be expected that the general principle would manifest itself—that the unit consumption would increase because of the decrease in total degree days. But in only three buildings was there an actual increase; in the other five buildings there was actually a decrease, although in one case the decrease was quite small. Nevertheless, it is obvious that the general tendency has been reversed.

Comparison of the Three Seasons

To establish some sort of quantitative relationship between the first year record and that for the second year, let us compare the total gallons and the total degree days. From the first year to the second, the total gallons consumed decreased by 29,281 while the degree days diminished by 688. The unit consumption increased by 2.4 gal per degree day. Expressed in percentage, the degree days went down by 10.38 per cent, the consumption fell off by 6.69 per cent, and the unit consumption increased by 3.6 per cent. Comparing the first year with the third, it is observed that the degree days diminished by 1026 or 15.5 per cent, the gallons decreased by 61,931 or 14.2 per cent, and the unit rate of fuel consumption increased by 0.9 or 1.36 per cent.

Using the first and second seasons as the basis for deviation in gallons per degree day with change in total degree days, a simple ratio can be established in this manner:

$$\frac{\%\text{ dd drop second year}}{\%\text{ dd drop third year}} = \frac{\%\text{ gal per dd increase second year}}{\%\text{ gal per dd increase third year}}$$

$$\frac{10.38}{15.5} = \frac{3.6}{x}, \text{ whence } x = 5.38 \text{ per cent}$$

From this it would seem that whereas the theoretical increase in unit consumption should have been 5.38 per cent, the actual increase was only 1.36 per cent. The difference of 4.02 per cent can be attributed to the benefit of the supervisory service.

A similar calculation can be made, as a check, using the ratios of the percentage change in gallons consumption for the basis:

$$\frac{\%\text{decrease in gallons second year}}{\%\text{ decrease in deg days second year}} = \frac{\%\text{ decrease in gallons third year}}{\%\text{ decrease in deg days third year}}$$

$$\frac{6.69}{10.38} = \frac{x}{15.5}, \text{ whence } x = 10 \text{ per cent}$$

From this point of view, since there should have been a theoretical drop of fuel consumption of but 10 per cent, and the actual decrease was 14.2 per cent, again the difference, or 4.2 per cent, can be credited to supervision.

It is thus established that the fuel saving the third season lies somewhere between 4.02 and 4.2 per cent. Probably the average of the two, or 4.11 per cent, represents the percentage of fuel saving during the third season. In gallons, this would be 0.0411 × 438,012 or 18,000, and since the average price paid per gallon was 4.27c, the cash saving amounted to $769.

It might be asked why there is even a slight discrepancy in the calculation of the savings by two different methods, since fundamentally they are based upon common data. The explanation probably lies in the validity of the two assumptions that were made in the calculations, namely, that variations in percentage change of degree days are directly proportional to (in the one case) change in unit consumption, and (in the second case) total change in consumption. It is unlikely that a chart of these variables covering a long period of time would be a straight line. Such straight line curves would have made the two calculated savings more nearly alike. Actually, by taking the average of the two calculations, a result very near the correct one is obtained.

Other Savings by Supervision

It is possible that there were other forms of saving in the operating expense of these eight buildings, beyond the fuel reductions already discussed. The item of maintenance, for instance, represents a definite expense to a building. During the heating season there arises normally some mechanical difficulty in the burner operation requiring the services of a mechanic. Practically all of that was eliminated by the regular inspection and adjustment service on the burners rendered by the supervisory service. Less tangible is the possible reduction in wear and subsequent replacement of burner parts because of the regular cleaning and check on lubrication.

It is not the intention of this article to imply in any way that the regular operating staffs at these buildings were not fully competent to fulfill their regular duties completely. In fact, judging by the manner in which coöperation was extended to the service engineers, the regular men were possibly better than average. The reason that the savings were made possible is perhaps simply that of the specialized training and special equipment possessed by the service engineers.

Life Expectancy of Gas Fired Heaters

Since preparation of the heating bulletin of the United States Housing Authority, the first part of which was published in the June HPAC, the USHA has raised the life expectancy of gas fired forced warm air heaters from 20 to 25 years. Consequently, the following changes should be made in section *D* of Table 4, published on p. 363 of the June HPAC. In the line "Forced Warm Air (Gas Fired)" the figure 0.0412 should be changed to 0.0312, and the figure 20 should be changed to 25.

Opposite

Gallons of fuel oil per degree day (horizontal scales) plotted against degree days per month (vertical scales) for Buildings Nos. 2 and 7 for the 1935-36, 1936-37 and 1937-38 heating seasons. The letters "S," "O," etc., indicate the months, starting with September and reading up

Ten Years Progress in Air Conditioning

By Walter L. Fleisher*

TEN years is a short time in the world's development but a long period in a young industry. In re-reading the article I wrote for the May, 1929, HEATING, PIPING AND AIR CONDITIONING, I find that I described the development of air conditioning in industry from its effect on goods, and some of the basic principles on which all air conditioning is predicated.

Comfort Air Conditioning Dominates

The last ten years have brought about an enormous amount of interest and a vast potential field for air conditioning for human comfort. In 1929, air conditioning for comfort, except in theaters, was a wished for, but seldom obtained, addition. Today, air conditioning for comfort is the dominant factor in the industry. The intelligent and experienced application of the accumulated knowledge we have of air conditioning promises to affect the common well-being and our energies to a major extent. Although I do not intend to imply that the health aspect of environmental conditions is not of long investigation, I do want to bring out the fact that the investigations which have been carried out and are being pursued with increased intensity in connection with health and comfort have extended the realms of air conditioning through many of the other basic sciences. Where 10 years ago we were simply a mechanical adjunct to any building or system, today we are the partners of the medical, the biological, the chemical, the electrical and the bacteriological professions and claim in each an important part. Our investigations of the effect of producing so-called comfort conditions have also spread into the industrial field. Where 10 years ago we discussed air conditioning in industry primarily with reference to the effects upon products and production, today we are weighing and comparing the conditions required for goods against the physiological reactions of the human beings who must work in this environment. With cheaper refrigeration and with a more general knowledge of the effect of better conditions on the output of human beings, we are aiding air conditioning in industry to a much greater extent than we ever have before.

Appreciating the results of these better conditions and realizing today that in many instances refrigeration is a necessary adjunct, we are searching for other means of producing the same results which may even be cheaper than the reduced price of refrigeration which today exists. Chemical dehumidification and colloidal dehumidification, combined with refrigeration or evaporative cooling, are making slow but decided strides forward.

Public contact with air conditioning increases with each year's new installations and where such applications bring added comfort, new and varied demands arise—possibly not as rapidly as many people now in the industry would like, but surely and promisingly. With the growth of general demand for air conditioning some inadequate and improper apparatus and design is creeping into the field. More young people, as well as older people, are being persuaded to enter the field as engineers, manufacturers or salesmen, and necessarily many of these will be disappointed in the results. However, all of us who have been in it for many years must recognize that air conditioning is no longer an exclusive province for the pioneers. It is a well established business with infinite possibilities, but, like all great developments, notwithstanding its possibilities only the better equipped can survive.

With all of these developments, with all of the money that has been spent on advertising air conditioning, on exploiting air conditioning and on research in connection with air conditioning, it has hardly touched its principal and most profitable field—the home. However, it is almost inevitable that in another 10 years air conditioning in the home will be a principal source of income to those devoted to its development.

Billion Dollar Industry

Ten years ago I prophesied that air conditioning would be a billion dollar industry. I still believe that this prophecy will be fulfilled, but only when the engineer, the manufacturer and the contractor have realized that every phase of air conditioning is worthy of study and development and that no one particular outlet should attempt to usurp the field.

Finally, the greatest progress will be made through the concerted effort of engineers, contractors, manufacturers and architects to avoid the promotion of inadequate equipment and the misapplication of good equipment. The public cannot be long deceived by artistically painted claims of comfort and quality of air where such do not exist. The market for air conditioning will not be advanced by the merchandising of gadgets under false claims.

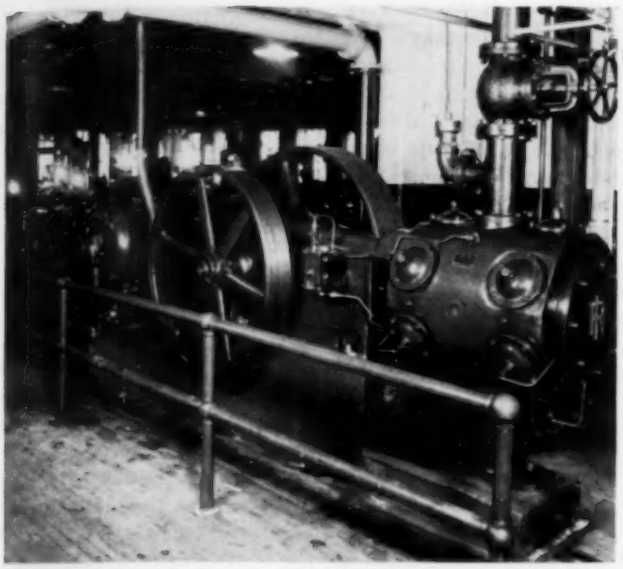
How a Pioneer Skyscraper Has Obtained Heat Economy

It appears that one of the principles involved in the heat switching machine described in the article of the above title published in the March 1939 HPAC is covered in an application for a basic patent by Oscar Ross which was made in 1925. Through subsequent steps patents Nos. 2,030,542, 2,030,543 and 2,030,544 were finally granted in 1936. It appears therefore that the machine described in the article, which was built and installed in the summer of 1931, may have been an infringement and in view of these facts the machine as described in the article has since been licensed under the Ross patents.

*Consulting Engineer. Member of Board of Consulting and Contributing Editors.

New Safety Code Tells How to Operate and Maintain Compressed Air Systems

By Dan L. Royer*



A motor driven compressor installation. Note double guard rail for wheel protection which extends around the machine and also protects the motor, motor pulley and compressor belt wheel. Note adequate working space around machine and drive

THE sections of the new ASA Safety Code for Compressed Air Machinery and Equipment dealing with installation requirements were described in the May HPAC. As the proper maintenance and operation of equipment have a great deal to do with safety, the sections applying to maintenance and operation are discussed this month.

Overpressure has been the cause of serious accidents in connection with compressed air equipment. It results from a derangement of control equipment. Rule 4.1, protective devices, is intended to assure proper protection against overpressure. It is evident that speed governors, unloaders and automatic pressure cut-outs, when installed, must be maintained so that their operation is reliable. After a great deal of consideration, the committee that developed the code briefly outlined the maintenance requirements for this equipment under rule 4.6, but in paragraph *d* of rule 4.1 required that electrically operated pressure cut-outs be so designed that contacts cannot fuse or lock in the running position. This was considered necessary because in spite of the best maintenance the fusing of such contacts in the running position has caused serious accidents.

Rule 4.3 requires that belts, pulleys, gears, flywheels and shafts be guarded in accordance with the mechanical power transmission apparatus code. The moving parts of compressors and their drives must be guarded to prevent injury to attendants and others who may be in the vicinity of the machine. It should be noted that guards should be arranged so that the compressor can be turned over by hand or with a jack, as this becomes necessary often enough to make solid guards without removable or hinged sections objectionable.

One of the causes of compressed air equipment accidents is the ignition of vaporized lubricating oil in the receiver or system. The committee found it impossible to make specific recommendations for lubricating oil in rule 4.4, but it was also found that the producers of lubricating oil have specifications for compressor oil that are entirely satisfactory. Under no circumstances should an air compressor be lubricated with an unsuitable oil. The relatively high temperatures in air compressor cylinders and discharge lines are sufficient to ignite the vapor that develops from the use of an unsatisfactory oil. It is also important to use *only* enough oil to lubricate compressor cylinders satisfactorily. To accomplish this, it is believed that visible flow, forced feed lubricators are necessary. These requirements are suggested in notes *A* and *B*.

Leaky compressor valves are responsible for increased compressing temperatures that not only result in inefficiency but may reach dangerous intensities. In paragraph *a* of rule 4.6, operation and maintenance, the regular inspection of valves is required and the correction of faulty valves by repair or replacement is made a mandatory requirement. Aside from the economic necessity for good valve maintenance, it is believed that this rule is of outstanding importance from a safety standpoint. The heating of the air in compressor cylinders resulting from churning caused by faulty valves has been the cause of receiver and discharge line explosions. Paragraph *c* of rule 4.6 ties in with paragraph *a*. High temperatures result in the carbonization of the lubricating oil and the impurities in the air that are being compressed. Various solutions are used to remove these deposits. Obviously, kerosene, gasoline or any other inflammable solvent should never be introduced into the compressor cylinders, receiver or discharge piping. The most practical method of eliminating these deposits is to use a mixture of soft soap and water.

Paragraphs *b*, *c* and *d of* rule 4.6 contain routine requirements for the tagging of throttle valves and line

Proper operation and maintenance of compressed air equipment and piping have a great deal to do with safety, and consequently are given special attention in the new ASA safety code covering such equipment. . . . Mr. Royer, chairman of the code committee, discusses operation and maintenance provisions of the code

*Chief Engineer, Ocean Accident and Guarantee Corp., Ltd. Chairman, American Standards Association Committee on Compressed Air Equipment and Machinery.

switches when compressor repairs are attempted, and for the regular inspection and test of unloaders, governor controls and other automatic control devices. These requirements are too frequently forgotten, although practically every person accustomed to the use of machinery is familiar with them. The necessity of seeing that these requirements are enforced is so evident that it should need no discussion.

Desirability of Aftercoolers

Section 5 suggests the desirability of aftercoolers, but does not make their installation mandatory. The use of aftercoolers in compressed air systems cuts down many operating and maintenance difficulties. A compressed air system without an aftercooler, when it is properly maintained and operated, certainly is not dangerous; it would therefore be impossible to require the installation of aftercoolers in a safety code. It is evident, however, that the installation of aftercoolers makes a safer system and increases the efficiency of the system operation. Notes *A*, *B*, *C* and *D*, rule 5.1, briefly point out these factors and encourage the use of aftercoolers.

The reduction in discharge air temperature after the air has passed through an efficient aftercooler is sufficient to cause the condensation of practically all moisture and oil vapor, so that its removal at the aftercooler is possible. When condensed oil vapor is removed at the aftercooler, the coating of the internal surfaces of receivers and piping systems with oily deposits is prevented, and temperatures are reduced to a point where the ignition of any remaining oil vapors is impossible. Aftercooling practically eliminates danger of the explosion of oil vapors in the discharge side of the system.

The elimination of moisture resulting from aftercooling prevents the freezing up of systems in cold weather, and the possibility of water hammer caused by accumulations of condensation at low points in the piping system. Strains in pipe lines are materially reduced by aftercooling, because pipe lines carrying uncooled air are continually expanding and contracting as the air flow varies. Such movement strains connections and produces leaks at joints.

From the standpoint of efficiency, aftercooling greatly improves the operation of utilization equipment. The condensation in uncooled air washes away the lubrication on air operated tools and machines quite rapidly, with the result that excessive wear resulting in high maintenance cost, accompanied by decreased efficiency, follows. Small quantities of lubricating oil are carried over from the compressor when the air is not cooled. This oil will coat the internal surfaces of rubber hose. As rubber exposed to oil deteriorates very rapidly, oil carried over from the compressor should be avoided if possible. These difficulties are eliminated by aftercooling.

Section 7, receivers, contains no provisions for the maintenance of receivers. Rule 7.2, installation, requires that receivers shall not be buried underground or situated in an inaccessible place. It was thought that it was unnecessary to stress the importance of keeping receivers free from external and internal corrosion. The fact remains, however, that the reduction in strength of receiver shells and heads, resulting from corrosion following exposure to dampness because of improper location, has been the cause of explosions. A receiver installed in accordance with the requirements of rule 7.2 should not require attention for a considerable length of time. If the external surfaces of the receiver have been protected by properly applied paint, there should be little difficulty with external corrosion. For this reason, the requirement "air receivers should be supported with sufficient clearance to permit complete external inspection and avoid corrosion of external surfaces" is of importance from a maintenance standpoint.

Rule 7.3, drains and traps, provides not only for the installation of a drain line properly valved but also requires that the receiver shall be completely drained at frequent intervals to prevent excessive accumulations of oil and water. Automatic traps for draining receivers are permitted and should be encouraged. The draining of air receivers should be a matter of routine, and it should be the duty of some responsible person to be sure that the receiver is drained regularly. While the installation of traps for draining receivers is desirable, the operation of such traps should be checked frequently to see that they are functioning satisfactorily.

Hose and Tools

The various kinds of pneumatic tools used in industry are too numerous to mention. While compressed air is used for many processes and in connection with equipment that permits of permanent pipe connections, there are numerous applications that require the use of rubber hose to conduct the compressed air to the final utilization equipment. Rubber hose eventually will wear out due to abrasion, kinking, and exposure to oil and gasoline. Persons using compressed air equipment are exposed to serious danger when hose, hose connections or clamps fail. For example, a workman was knocked off a steel structure, with fatal results, by the sudden movement of the air hose to a riveting hammer when a hose connector some distance from the tool failed without warning. Rule 8.1 requires that hose and hose connectors be designed for the pressure and service to which they are subjected. It is recommended that hose connectors and clamps be of the heavy duty type. The installa-

Damage resulting from an air receiver explosion. Seams welded in a faulty manner failed when a motor control contact fused in the running position and the safety valve failed to operate

tion of hose clamps should be entrusted only to a competent person who is thoroughly familiar with the proper methods of installation of these devices. Under no circumstances should any "hay wire" expedients with hose connectors and clamps be employed.

Paragraph *c* in rule 8.2, portable tools, requires that no tool change or other work be attempted on any piece of hand operated utilization equipment unless the stop valve is closed. This rule is intended to prevent the practice of kinking the hose to make adjustments or do other work on hand tools. To shut off air by kinking the hose not only shortens the life of the hose, but is an unreliable and dangerous method that should not be tolerated.

Rule 9.1, working space, and rule 9.2, aisles, require the location of compressors and compressed air equipment to have sufficient space to give safe access to all parts of the equipment, and adjacent aisles to be adequate and not used for storage space or similar purposes. The undesirability and dangerous possibilities of cramped working spaces around compressed air equipment are worthy of consideration from a maintenance standpoint. For example, if a two cylinder compressor is provided with an intercooler, adequate space must be provided between an adjacent wall and the intercooler head so that intercooler tubes can be removed when necessary.

Paragraph *c* in rule 9.3, safety valves, requires that all safety valves be tested frequently and at regular intervals. The importance of this requirement is often forgotten. In normal operation, the safety valves protecting a compressed air system are seldom called upon to function. It is only when some unusual condition exists in the system that the pressures rise to a point where the safety valves are called upon to relieve. Safety valves (like any other equipment) must be tested to be sure that they are in satisfactory operating condition. This can be accomplished by pulling the test lever when the pressure on the system is at its normal point and observing the effort necessary to lift the valve disc from its seat at a given pressure.

It is hoped that the Safety Code for Compressed Air Machinery and Equipment will provide a guide for the many organizations that have occasion to install and operate such equipment. Safety is not attained even with properly installed equipment unless it is maintained and operated intelligently and in a manner that will prevent the development of unsatisfactory operating conditions. A safe installation is the first step toward the safe and economical operation of the system. After the installation has been made in a manner that meets all safety and efficiency requirements, satisfactory maintenance and operating methods must be followed.

[*Concluded from p.* 424]

could be established in percentages of the easily measured input to the heat transmitters. This comfort rating possibly would be knee-high dry bulb temperature measured at some given number of feet away from a heat transmitter placed below a window, or in case of panel heating, with the panel in the ceiling. Thus the A company might be allowed to rate its "No. 27" convector at 9.6 Mb† input, comfort factor 110 per cent, so that the unit instead of being a conventional 40 EDR one would be rated at 10.56 Mb.

Another device perhaps of the same weight and size made by the B company on duplicate test with 9.6 Mb input would show a comfort factor of 80 per cent and would be comfort-rated at 7.68 Mb.

I proposed a scheme like this when chairman of the ASHVE committee on research before we had even the present codes for testing by input. I may be over sanguine but at least progress has been made. The rating has a remote resemblance to power factor in alternating current electric practice.—Samuel R. Lewis.*

Air Conditioning the New Mauretania

The air conditioning plant installed aboard the T.S.S. *Mauretania* includes the features of modern land installations, the apparatus being constructed to meet the heavy duty requirements of marine service.

The installation has been designed to provide comfortable conditions of temperature and humidity in the principal public rooms under all weather conditions likely to be met with in Atlantic or cruising service. In hot weather temperature reductions up to 15 F will be possible, while in very warm humid weather the relative humidities will be as much as 35 per cent lower within the rooms served by the installation than outside.

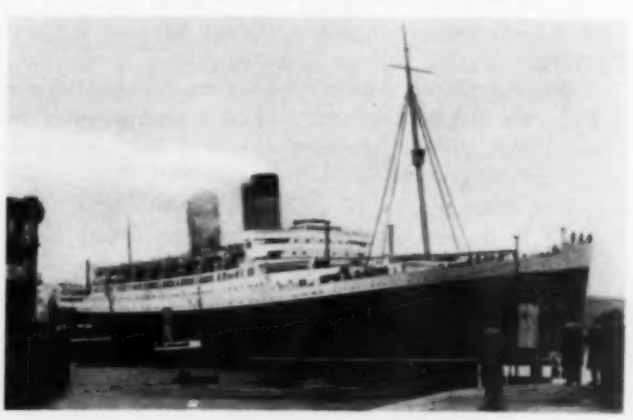
A minimum winter temperature of 10 F was the basis of the heating and humidifying design, and the plant will be capable of maintaining inside temperatures of 70 F with a relative humidity of 45 per cent in such weather.

The cabin dining saloon is served by two plants of equal capacity comprising supply fan, dehumidifier, steam heater batteries, automatic control dampers, while in the other rooms in which the equipment has been installed there is a single equipment thus providing each room with independent control. These plants have a combined capacity of approximately 3,500,000 cfh of air.

The refrigeration is supplied from CO_2 machines associated with the ship's main refrigerating machinery and is distributed by means of cold water which is sprayed in the various dehumidifiers.

The automatic control system is pneumatic. Although the various air conditioning plants differ in arrangement, each comprises essentially a supply fan, marine type dehumidifier, preheater and main heater, and automatically operated air dampers.

The new Cunard White Star Liner *Mauretania*



*Consulting Mechanical Engineer. Member of Board of Consulting and Contributing Editors.
†Mb = 1000 Btu.

USHA Gives Suggestions for Design of Heating Systems

RECOMMENDATIONS and suggestions for the design of heating plants are presented herein. The United States Housing Authority recognizes that there are various satisfactory methods of handling particular design features, and these suggestions are not to be considered the only acceptable ones.

A. The Central Plant

1) *Boilers*

Central plants: design to operate at 80 to 100 lb pressure, with not less than three boilers; water tube boilers, at 150 per cent of rating; portable fire box boilers, at 100 per cent of rating. Normal rating should not exceed 500 hp except where more than four boilers are required.

Boilers up to 100 hp: fire box type. Boilers 100 hp to 300 hp: either straight water tube, or low head three drum bent tube. Boilers above 300 hp: either straight water tube, or high head four drum bent tube.

Insulation of settings on coal burning boilers 300 hp and over will be improved by providing air cooled walls. For oil or gas fuel, settings may have solid walls and air cooled floors. For straight tube boilers, provide solid end and side walls 22 in. thick with full floating bridge wall to take care of expansion. Provide for expansion in the setting walls as well as proper clearance between drums and settings and supporting structural members. Support boilers independently of brick setting.

Provide for easy cleaning of all water and fire surfaces. Provide soot blowers on water tube, horizontal return tubular, and portable type fire box boilers burning bituminous coal.

Include water treatment facilities where necessary.

Provide for easy removal of ashes; if pulverized coal firing is used, provide a method for trapping and removal of fly ash.

A clear space of 14 ft 0 in. minimum is recommended between boiler fronts and the wall of the building, not less than 6 ft 0 in. between the rear of the boilers and the wall of the building, and not less than 6 ft 0 in. between each boiler for air cooled walls and 5 ft 0 in. for all others. Each boiler should have an individual setting.

Provide proper ventilation over the top of all boilers. This can best be accomplished by a monitor over each boiler with pivoted sash and extended operating device.

2) *Steam Outlets*

The steam outlet of each boiler carrying in excess of 15 lb gage pressure should have an automatic stop and check valve at the boiler, and globe or angle valve at the header. One or more openings for future flanged connections may be provided on boiler drums.

3) *Cat Walks*

Provide cat walks with ladders so that all points on boilers which must be serviced may be reached.

4) *Stacks*

Base peak boiler loads on a chosen minimum outside temperature.

Stack diameters should be the most economical size, calculated for peak load demand at 20 F above design temperature with provisions for further extension if contemplated.

Design for 50 per cent excess air under peak load demand for oil and 90 per cent excess air for coal.

At least 0.2 in. of draft is needed in the combustion chamber for forced draft equipment when burning either coal or oil.

Specific recommendations for natural draft fuel oil burning equipment will be furnished by the United States Housing Authority on request.

Where either oil or gas is used, it is advisable to calculate the stack size on the basis of burning coal with and without cinder trap.

5) *Overhead Bunkers*

Overhead coal bunkers in central plants: design with a capacity of not less than 10 per cent of the estimated maximum annual coal consumption.

Where more storage is desirable, provide outside storage spaces for an additional 5 to 10 per cent. These may be surrounded by a masonry wall at least 6 ft 0 in. high.

In localities where the analysis of the coal shows a content of 3 per cent or more of sulphur, use overhead bunkers either of concrete or metal coated on the inside with at least ⅛ in. of a bitumastic paint or a tile lining set in hot mastic.

6) *Underground Coal Storage*

Where underground storage is provided, the coal holes should be raised about 4 in. above the trucking grade. They should be sufficient in number and so located that at least two-thirds of the total volume of storage can be filled without hand trimming. Openings from these underground storage bins to the boiler rooms should have a metal door, to be closed when coal is being placed in the bin.

Underground storage is recommended only where the total annual consumption is less than 3500 tons.

7) *Coal Handling Equipment*

Traveling weigh larries, electrically operated, should have capacities of not less than 1500 lb for boilers up to 250 hp, and 2000 lb for boilers above 250 hp, and should be equipped with switches to limit the horizontal travel. Electric motors for weigh larries and other coal handling equipment should preferably be the fully enclosed, fan cooled type. The use of this equipment ought to be limited to projects where the total annual coal consumption exceeds 3500 tons.

Coal hoppers for stoker feeds: design for 1000 lb for boilers up to 250 hp, and 1500 lb for boilers over 250 hp. Stokers may be equipped with fully enclosed, fan cooled electric motors or steam turbines, or may be hydraulically operated.

Monorails may be supported from roof trusses of boiler rooms, but should be placed so that the bottom of bucket, when raised to its unloading position, will not be over 18 in. above the top of the stoker hopper, to limit the swinging of the bucket.

Hoists should have electric lift and travel, and two lifting cables placed on either side of the bucket to prevent the bucket from turning when being raised.

The bucket should have at least 1000 lb capacity with a side bottom dump and gate of the clam shell type operated with a wheel, rack and pinion. The wheel should operate through a knuckle joint so that it can be reached when bucket is raised or lowered. Weighing sections should be placed in the monorails and the longitudinal travel of buckets limited to 50 fpm maximum.

Drums of hoist should be grooved for rope and not flat.

8) *Ash Handling Equipment*

When the annual coal consumption is up to 2200 tons, consideration must be given to the method of ash disposal.

When the annual coal consumption is above 2200 tons, a pneumatic or preferably a mechanical system of ash handling is desirable.

Where cinder traps are used, provide for disposal of fly ash through the ash handling system. Dispose of the fly ash from the rear hopper of the boiler through the ash handling system, or return into the boiler furnace for burning the combustibles.

9) *Instrument Boards and Instruments*

The following instrument boards are recommended for heating plants burning in excess of 2500 tons annually:

a) One main instrument board located in the boiler room as directed. Instruments on this board may include:

1) Electric clock, 6 in. diameter.
2) Smoke density indicator and recorder where required.
3) Recording and integrating steam flow meter, for yard distribution system only.
4) Steam pressure gage indicating boiler pressure.
5) Steam pressure gage indicating yard distribution system.

b) An instrument board for each boiler, located adjacent to the boiler as directed. Instruments on this board may include:

1) CO_2 recorder.
2) Exit gas recording pyrometer.
3) Indicating and integrating steam flow meter.
4) Steam pressure gage furnished under boiler trimmings.
5) Draft gage that will comply with the draft requirements by having the requisite number of pointers.

10) *Draft Control*

Control of each boiler should be obtained independently through its own uptake damper.

Arrange main stack damper for hand control with locking quadrant.

11) *Test Openings*

It is well to provide insertion points in breeching at stack for pyrometers for taking test temperatures.

12) *Feed Water Heaters*

Use feed water heaters of the deaerating type, operating at not less than 215 F, with a reserve storage capacity of not less than 5 min between overflow and cold water make-up inlet.

The second part of the United States Housing Authority's bulletin on the heating of low rent housing projects is given here; it covers recommendations and suggestions for designing the heating system. . . . The introductory paragraphs of the bulletin, and Part 1 on economic analysis, were printed on pp. 361-364 of the June HEATING, PIPING AND AIR CONDITIONING, with forms for the economic heating analysis.

13) *Condensate Receiving Tank*

In all cases a condensate receiving or surge tank should be used having a capacity of 30 min storage between overflow and automatic float controlling make-up water.

Where space conditions will not permit the use of a receiving tank, the feed water heater should have not less than 20 min storage capacity between the overflow and float valve make-up.

A cold water bypass should be arranged on the boiler feed pump suction, for the purposes of boiler wash, for reducing feed water temperatures, and for the operation of hydraulic tube cleaners. The pumps should be arranged to maintain feed water service to any boiler, while another boiler is being given hydraulic service.

14) *Boiler Feed Pumps for Central Plants*

With installed boiler horsepower up to 400 hp, two pumps are recommended. With installed boiler horsepower from 400 to 1200 hp, three pumps are recommended. With installed boiler horsepower over 1200 hp, special consideration should be given to the number of pumps.

No pump should have a capacity greater than 75 per cent of the peak demand and all pumps should be steam driven.

When three or more pumps are installed, one should be sized to take care of the summer load when operating normally.

Valve all equipment furnished in duplicate so that either or both may be operated singly or together.

15) *Primary Pressure Reducing Valves*

Mount the primary pressure reducing valve assembly in a horizontal position, on one of the side walls, rather than in a horizontal position near a ceiling. Provide platforms for servicing.

Experience has shown that three or more pressure reducing valves, arranged so that one valve can take care of the entire hot water load in the summer, provide a more flexible installation. As a basis for calculation, the following may be used as a guide:

Assume a heating load of 12,520 lb of steam per hr, a hot water load of 4550 lb of steam per hr, equal to 17,070 lb of steam per hr plus 10 per cent for line losses—a total of 18,777 lb of steam per hr.

The pressure reducing valve to take care of the hot water load should be adequate for 4550 lb of steam per hr, plus 10 per cent line loss—a total of 5000 lb of steam per hr.

Subtracting this from 18,777 lb of steam per hr, leaves 13,777 lb per hr to be taken care of by two pressure reducing valves, one valve to take care of 75 per cent of this and the other to take care of the balance.

By this arrangement, the valves may be opened or closed to meet the increase or decrease of the demand steam load. In no case should these valves be larger than 8 in.

Protect all pressure reducing valve stations by a battery of pop safety valves having a capacity of *60 per cent* of the steam demand load.

B. The Group Plant

Group plants are usually preferable to individual building plants, although the latter may be practical for very large buildings. Design recommendations for group plants are generally applicable to individual building plants. Group plants should be located to allow economical interconnection to insure flexibility under mild weather and summer load operation.

Boilers in these plants should be of the portable fire box type sized at 100 per cent of rated capacity. Each plant should have at least two boilers, with the capacity of each boiler not to exceed 12,000 sq ft of direct radiation.

1) *Piping at Group Boiler Plants*

For single or two pipe gravity steam heating systems, equip boilers with regular Hartford loops, and valve so that any or all boilers may be operated or drained without interruption of service.

2) *Boiler Feed Pumps for Group Plants*

Whenever possible, arrange group plants carrying up to 15 lb gage pressure so that the vacuum pumps can discharge water of condensation directly to the boilers. Where this is impractical, use single stage motor driven boiler feed pumps.

3) *Steam Outlets on Boilers for Group Plants*

Equip steam outlet of each boiler carrying up to 15 lb gage pressure with one globe valve in each connection.

4) *Motor Truck Scales*

Where it is not possible to secure certified weights of coal deliveries, include motor truck scales of the beam type, installed in pits, if feasible. The scales should have a capacity of not less than 70,000 lb.

C. The Distribution Lines

The design of the heating distribution system should not be deferred until after the site plan has become set. Both the character and extent of the distribution system may be influenced by close coöperation between the heating engineer and the site planner from the beginning. Buildings arranged compactly will achieve economy in heating distribution. Where plenty of land is available, it is more economical to concentrate the area not needed for housing in a single large unit than to space the buildings widely over the entire site.

Underground steam and return distribution should be direct: design layouts to take care of expansion at the point of entrance into each building, where possible.

Reduce the number of steam drip traps located in yard steam manholes to a minimum, and wherever possible provide for dripping yard steam mains by traps placed inside the nearest basement.

Where pipe expansion must be provided for underground lines, use welding fittings and straight pipe expansion bends, installed in minimum design non-accessible expansion chamber.

Underground piping should be installed in a simple type of conduit, consisting either of split tile, built-up tile, pre-cast lightweight concrete with a continuous concrete base and gravel backing where required, sponge-felt asbestos pipe covering with integral waterproof jacket, or of a combination casing and covering conduit constructed of standard thickness 85 per cent magnesia pipe covering without canvas wrap, sealed into an inner core by application of a uniformly thick layer of high melting point asphalt between the outer surface of the magnesia insulation and the outside metal jacket.

Any and all other types of conduit of equal or lesser cost should be given full consideration. All conduit installations should be specified to be substantially in accordance with the manufacturers' commercial standards.

D. Heating Within the Buildings

1) *Radiation and Risers*

Supply and return risers: expose and arrange to serve two radiators on each floor wherever possible.

Arrange connections for both supply and return on same end of radiator, nearest to risers.

Install radiator supply and return branches above floor in fireproof floor construction.

Install radiator supply branches above floor, in wood joist floor construction, and return branches under flooring for stability. Floor joists may be notched near bearing for this purpose. All connections through floors, walls or partitions should have sleeves and escutcheons.

The amount of radiation for each radiator called for on the riser diagram or schedule should be the minimum required.

No radiator should be installed which does not provide at least the amount of radiation called for on riser diagram or schedule. This is to be based on the manufacturers' catalog ratings for cast iron, except that the sizes given may vary slightly to suit manufacturers' standards, provided the total radiation specified for any one individual dwelling unit is not decreased.

Wherever possible, use exposed risers for heating kitchens and bathrooms.

E. Drying Room Heating

Construct clothes drying rooms in groups adjacent to laundries, locate as close as practical to hot water generator rooms, and equip with unit heaters delivering 120 deg air and supplied either with steam or with hot water from the domestic hot water supply system.

When steam is used, the supply should be taken from the hot side of the zone control valve so that heat may be available during the summer season. When hot water is used, the unit heaters may be supplied from the basement hot water mains.

Drying rooms should have exhaust fans capable of exhausting at least six changes per hour.

When laundries are adjacent to drying rooms, the exhaust fans may get the majority of their supply through the laundries. Insulate the ceiling of all drying rooms to protect the rooms above from the heat of the drying rooms.

Consideration should be given to the installation of cabinet type drying units in lieu of drying rooms, as substantial reductions in cost of such equipment have recently been effected through redesign. This type of dryer, using either gas or electricity, requires much less basement space, and substantially increases available laundry facilities through more rapid clothes drying.

[To be continued]

Air Conditioning Load Factors Are Improved by Storing Refrigeration

By D. C. Pfeiffer*

ANALYSIS of air conditioning installations in Dallas, Texas, has indicated that the average annual consumption is 844 kwhr per connected hp for office buildings and 1132 for department stores. A further analysis indicates an average load of 0.8 kw per connected hp, recorded by the 15 min demand meters. On this basis the office buildings average 1055 kwhr annually per kw of customers' demand and the department stores 1415 kwhr per kw. This corresponds to an annual load factor of approximately 12 per cent for the office buildings and 16 per cent for the department stores.

Types of Storage Systems

One method of improving the air conditioning load factor is to provide a means of storing refrigeration at night when the normal building load is low and using the "stored" refrigeration to supplement the machine capacity during the day. There are three general types of storage systems:

Sweet Water System

For the sweet water system, a large water storage tank is used. At night, the water in this tank is cooled to about 36 F by the refrigerating system. During the day when the air conditioning system is in operation, the water which has been cooled on off peak operation is injected into the chilled water circulating system. When water is injected into this system, an equal amount of water is displaced through piping into the opposite end of the tank from which the cold water is being taken. The tank is provided with partitions or baffles to prevent mixing of the warm and cold water.

If the water is cooled to 36 F and the conditioning units so designed that water returns from them at 58 F, water drawn from the tank is thus warmed 22 F in passing through the air conditioning system, and each cubic foot will extract 1372 Btu from the system. A ton of refrigeration is defined as the removal of heat at a rate of 12,000 Btu per hr; consequently, it is apparent that approximately 10 cu ft of water is the equivalent of 1 ton-hour of refrigeration, allowing approximately 10 per cent for losses in the tank. Thus, for a 10 hr operating day, 100 cu ft of tank capacity is necessary to replace each ton of refrigeration equipment capacity. If the operating time of the air conditioning system were but 5 hr each day, then 50 cu ft would replace the operation of 1 ton of refrigerating equipment.

Brine System

Tanks containing brine or an anti-freeze solution have been used in some places. The advantage of this is that the brine can be cooled below 32 F; consequently a greater storage of refrigeration can be accomplished per unit of volume provided. If the brine is cooled to 14 F instead of 36 F (as mentioned above), then the available temperature range has been increased from 22 F to 44 F, or doubled. Doubling the temperature range will double the effective capacity of the tank; consequently a tank one-half the size could be used as compared with the sweet water system.

The brine system has the disadvantage of requiring an additional heat transfer between the brine and the water circulated in the air conditioning system. It also has the disadvantage of higher operating costs due to the lower operating suction pressure of the compressor.

Ice System

The ice system has been used very successfully in connection with air conditioning. To freeze a pound of water into ice, 144 Btu must be removed; this is known as the heat of fusion of ice, which becomes available for cooling water when ice is melted. This amounts to approximately 9000 Btu for each cu ft of water and, in addition, refrigeration is available by warming the water from 32 F to 58 F when circulated through the air conditioning system. This amounts to more than 10,600 Btu per cu ft of water or approximately 7½ times as much as with the sweet water system.

Although it may appear that the tank size could be reduced to ⅐ of that required for the sweet water system, this is not true, since space must be allowed for refrigeration piping on which ice is formed and for channels through which water must be allowed to flow in order to melt the ice at the desired rate. The design of the ice-forming coils and tank requires a great deal of thought to obtain an economic balance between the lineal feet of coils and the ice thickness. As the ice thickness formed around the coil increases, it acts as an insulator, retarding rapid heat transfer from the refrigerant inside of the coil to the periphery where additional ice is being formed. With increasing ice

A method of improving the air conditioning power load factor is to provide a means of storing refrigerating effect at night when the normal building load is low and using the stored refrigeration to supplement the machine capacity during the day. Results of recent installations of this kind in Dallas indicate an annual load factor in excess of 20 per cent. . . . Mr. Pfeiffer described the types of storage systems, discussed the economic amount of storage, and reported operating results in a paper at the University of Texas air conditioning conference, from which the information here has been taken

*Sales Dept., Dallas Power & Light Co.

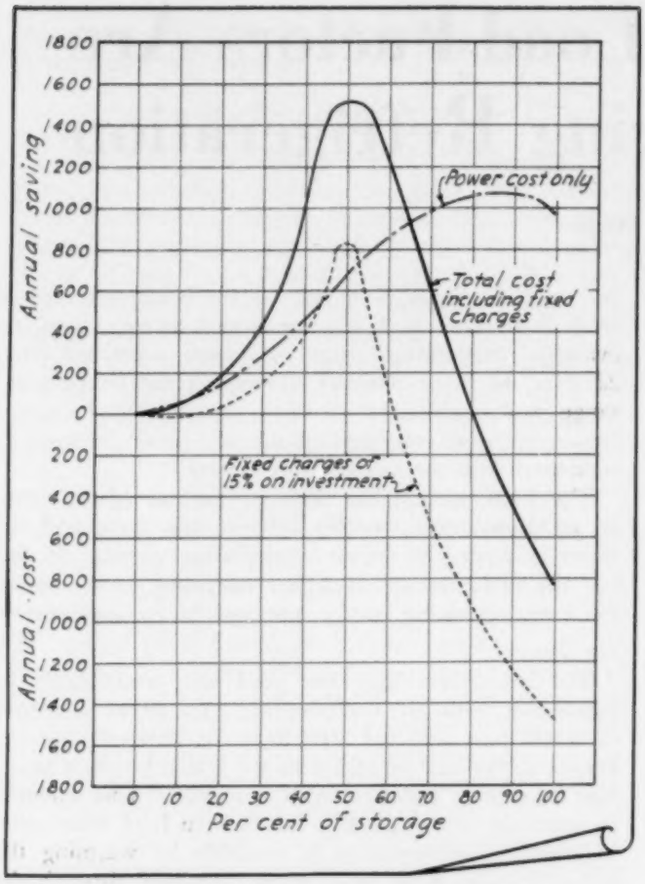


Fig. 1—Economical per cent of storage in one case.

thickness lower refrigerant temperatures must be maintained to continue the formation of ice at a given rate. A greater amount of cooling surface will allow thinner ice to be formed, which will result in a lower operating cost and also give a greater melting capacity. The design of such a tank should be undertaken only by an experienced and qualified engineer.

Economic Amount of Storage

In considering the application of storage to the air conditioning of a building there are a number of factors to be considered. First, the electric load curve of the building must be studied to determine the hours during a 24 hr day when the refrigeration equipment can be operated off peak. It is apparent that for an average office occupied 8 to 10 hr per day, insofar as the operating costs alone are concerned, the most economical arrangement would be to provide sufficient refrigerating capacity in compressors and sufficient storage capacity so that operation of the compressors would be only at night. In this case, the storage tank would carry the entire air conditioning load and the power cost would be largely a matter of added energy costs with no demand charge. This would be known as a 100 per cent storage system. Obviously the 100 per cent storage system would require at least the same capacity in refrigerating equipment as the conventional air conditioning system and, in addition, a storage tank of the same capacity as the compressors. Such a system would cost more than the conventional type by the amount of the storage tank and its necessary auxiliaries.

If, in the same building, compressors of one-half the capacity were installed to operate both day and night, and a tank were installed of a capacity equal to one-half of the daytime building cooling load, then the compressor capacity would be the minimum possible and this would be a 50 per cent storage system. In this case the operating costs would be greater than the 100 per cent storage system, as there would be a demand charge for the refrigerating system during the daytime of approximately one-half of that for the conventional, non-storage system.

It is apparent that somewhere between the conventional, non-storage system having sufficient compressor capacity for the load with no tank, and the 100 per cent storage system having the same compressor capacity and in addition a storage tank of equal capacity, there is an economic balance between machine capacity and tank capacity. This is indicated by Fig. 1, which was made for a typical Dallas office building, and the curves show that with the assumed electric rates, 50 per cent storage is the most economical, considering both fixed charges and operating costs. The curves show how the increased investment affects the per cent of storage which may be justified and it is obvious that any storage system which increases the investment materially over the conventional, non-storage system is difficult to justify.

Close coöperation between the utility power salesman and the air conditioning representative is necessary in determining the most economical per cent of storage to be used.

Operating Results of Recent Installation

The 19 story Medical Arts building at Dallas is in the shape of a maltese cross, having a high ratio of wall and window area to floor space. The 550 ton air conditioning installation serves the main portion of the building to the 16th floor; future extension for the 17th, 18th and 19th floors, including hospital quarters and auditorium therein, will require approximately 90 tons.

Refrigeration equipment is in the basement, occupying new space secured by excavating an originally unused area below street level. Cooling towers for the condensing water system are on the roof and make-up is obtained from city mains. Motor driven centrifugal compressors are of latest design with built-in condenser and evaporator; they have economizers involving provision for taking flash vapor into an intermediate compressor stage. The two stage air washer serving the entire installation is at an intermediate level, placing its basin water line about 26 ft above the overflow line in the storage tank.

Since excavation and concrete work between building column footings was necessitated in any event to accommodate refrigeration equipment, it proved comparatively inexpensive to excavate additional space and to construct a reinforced concrete tank for chilled water storage. Locations of existing column footings resulted in an L-shaped tank with overall dimensions 28 ft by 72 ft, from which an area 11 by 30 ft at one corner was eliminated. With a 12 ft inside depth, water holding volume is 20,000 cu ft. Cork insulation 3 in. thick is applied on the interior of the tank, waterproofed with bitumastic membrane. Internal baffles for preventing

the mixing of cold and warm water are placed vertically, spaced 5 to 6 ft apart throughout the entire length of tank. Alternate baffles extend above the working water line and terminate about 1.5 ft above the tank floor; intermediate baffles are kept about 1 ft below the water level, and extend to the tank bottom. Provision is made for venting air spaces between alternate baffles above the water line, and small openings are similarly provided at the bottom to permit flushing out sediment. Construction used for the baffles is stucco on metal lath or wood forms, attached to side wall cork insulation and to several intermediate poured concrete columns.

The motor schedule for the air conditioning system is as follows:

Initial Installation	Rating, Hp
Central dehumidifier fans	37
Booster fans (19)	50
Unit conditioner fans in 1st floor stores (4)	11.5
Exhaust fans (2)	4.5
Compressors (2)	500
Chilled water pumps (3)	75
Condensing water pumps (2)	60
Cooling tower fans (2)	20
Future Addition	
Booster fans (3)	6.5
Chilled water pump	15
	779.5

Compressor motors are rated at 200 and 300 hp, for machines with a total rating of 415 tons when producing 44 F water. Chilled water pumps needed for summer operation are rated 7.5, 27.5 and 40 hp, and for the future unit 15 hp. Each condensing water pump is rated at 30 hp, and the cooling tower fans 10 hp.

Allocation of refrigerating capacity is approximately as follows:

Main building, 2nd to 16th floors	295
Annex, 4th to 16th floors	195
First floor stores	60
Future hospital	70
Future auditorium	20
	640

The system was put into service during the latter part of the 1937 cooling season, but no effort was made to obtain representative information until 1938. For the purpose of assisting the building management in obtaining the most efficient operation, and also to obtain as much information as possible regarding this type of system, submeters were installed on various portions of the air conditioning electric load. These were arranged to meter separately the loads of the air handling equipment, the chilled water pumps, compressors, condensing pumps, cooling tower fans and also the total air conditioning load. Fig. 2 is a weekly demand chart for the air conditioning system and shows the night operation for recharging the storage tank.

Data for a typical summer month, July 26 to August 25, indicate that the total consumption for air conditioning was 219,600 kwhr, as compared with 396,496 total for the building. Of the total used for air conditioning, 178,400 were used by the compressors, condensing water pumps, and cooling tower fans, 28,920 were used by the air handling equipment, and 12,280 were used by the chilled water pumps. The total demand of the building for this month was 1104 kw, of which 585 was accounted for by the air conditioning system, and includes 468 kw chargeable to compressors, condensing water pumps, and cooling tower fans. Of the total demand of the air conditioning system, 89 kw was chargeable to the air handling equipment and 28 kw to the chilled water pumps. This gives a monthly load factor of 52 per cent for the air conditioning system, 53 per cent for the compressors, condensing water pumps, and cooling tower fans, 45 per cent for the air handling equipment, and 61 per cent for the chilled water pumps.

Actual records of eight months have been obtained from meter readings, and by estimating the remaining four months (which are largely non-cooling months), an approximation can be obtained of the total kwhr per kw of demand which we can expect for systems of this type. The eight months (June, 1938, to January, 1939, inclusive) show a total consumption of 1,063,400 kwhr, and it is estimated that the remaining four normally non-cooling months will use approximately 210,000 additional, making the estimated annual total air conditioning 1,273,400. The maximum air conditioning demand occurred in September—it was 614 kw, giving a value of 2074 kwhr per kw of maximum air conditioning demand, or an annual load factor of 23 per cent.

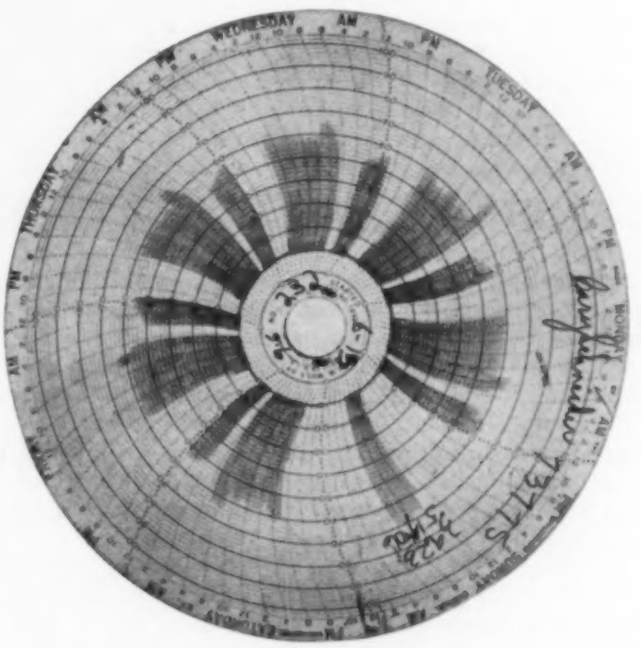


Fig. 2—Weekly demand chart for air conditioning system

When comparing these figures with conventional systems, consideration should be given to the kilowatt-hours used by the fans for winter conditioning in the Medical Arts building. It is estimated that approximately 20,000 kwhr per month for five heating months will represent the fan consumption. This 100,000 kwhr should be deducted from the total if the winter operation is not included. After deducting this from the total, the value of 2074 kwhr per kw of demand reduces to 1911. This correspondingly reduces the annual load factor of the air conditioning system from 23 to 21.8 per cent.

There is another consideration in making comparisons of this building to other office buildings in Dallas. Due to the nature of the occupancy of the Medical Arts building, its operating load factor other than air conditioning is somewhat higher than the average office building. In 1937 the load factor of this building was 45 per cent, which included very little air conditioning. This compares with an average of from 32 to 42 per cent for other office buildings and department stores. Because of this, it is believed that a further reduction of 10 per cent should be used in comparisons of expected results from storage systems. Reducing the value of 1911 kwhr per kw of demand by 10 per cent, this then becomes 1720 or an annual load factor of 19.6 per cent. This reduced air conditioning load factor, for comparative purposes, is still 63 per cent better than the average office building and 22 per cent better than the average department store. It is apparent from the comparison that increases in annual load factor from 25 to 50 per cent can be realized from this type of storage system, and the fact still remains that the actual annual load factor of the air conditioning system of the building for the 1938 season is slightly in excess of 23 per cent.

[A second article will discuss the storage system in the Dallas Power & Light Co. main office building.]

Refrigerating Engineers Hear Papers on Air Conditioning

IMPROVED operation of bus and auto engines through the use of clean, conditioned air in the carburetor was explained before the spring meeting of the American Society of Refrigerating Engineers at Hershey, Pa., May 21-23, by A. J. Mallinckrodt and Lars Hanson.* They outlined a method now in use on air conditioned buses whereby clean, filtered, dehumidified air is taken as a "byproduct" from the conditioned passenger space and run through the customary air cleaner on the carburetor. This results in more than double cleanliness and eliminates the need for constant cleaning of the air cleaner, particularly in regions where buses travel over unpaved roads. While the installation is too new for complete results, indications are that the life of the motor will be lengthened and its efficiency improved.

The paper pointed out that one additional passenger per trip more than pays the cost of an air conditioning system on a modern bus. In summer, un-air conditioned bus lines lose passenger traffic to air conditioned trains, particularly on southern routes.

Charles A. Peters† spoke on air conditioning in government buildings in Washington, D. C. Fully air conditioned buildings comprise approximately 11,000 tons of the 12,000 tons of refrigeration operated by the National Park Service; in addition to the buildings under the jurisdiction of the National Park Service, other air conditioned government buildings in Washington add some 7000 tons to this figure.

Mr. Peters discussed operating problems, with particular reference to condenser water, which comes from three sources: the city water supply, cooling towers, and the Potomac River. He commented briefly on the importance of air movement and control, the future of air conditioning in Washington, and on operating costs. He gave figures on the yearly cost of the new post office building air conditioning system as $0.0465 per gross sq ft of building area, $26.00 per rated ton, and $15.35 per occupant. These figures include only electric current, operating and maintenance labor, supplies, materials, repairs, and replacements.

A paper on the thermodynamic properties of methyl chloride, by H. G. Tanner, A. F. Benning and W. F. Mathewson**, presented a discussion of the subject based on a critical search of the literature and a set of equations utilizing available experimental data to express the desired thermodynamic properties. Final computations of tabular data from the precise equations were checked for arithmetical mistakes by the method of differences, and by duplicate calculations of individual points at random temperatures. As an added precaution against errors in computation, about 15 per cent of all the calculations were duplicated by persons working independently.

The tabulations presented gave the properties of saturated vapor and superheated vapor.

V. C. Patterson†† discussed refrigeration applications at the Hershey Chocolate Corp. plant, covering the growth of refrigeration there, the variety of refrigerating processes employed, and details of a cooling system recently installed for processing cocoa powder. The present plant consists of 1665 tons refrigerating capacity using ammonia, and 425 tons using "Freon-12."

The installation for processing cocoa powder comprises a two stage system with fully flooded evaporators and float controlled refrigerant feed, the first stage unit being built up of six banks of steel spiral finned coils arranged for gravity refrigerant feed from a common surge drum in which the level of ammonia liquid is controlled with a float valve, and the second stage unit comprising bare pipe trombone coils arranged in a tunnel with a float valve for maintaining a constant refrigerant level in the low pressure receiver. Liquid ammonia is pumped from this receiver into the bottom of the trombone coils, then up through the coils and back into the low pressure receiver, producing a fully flooded condition in all the coils. Advantages of the system were summarized as follows:

It is compact, the actual floor space occupied being approximately 25 per cent of that occupied by the original bunker room.

Part of the moisture load is carried by finned coils operating at high suction pressure, thus reducing the quantity of moisture to be frozen out on the bare pipe coils and thus reducing defrosting requirements.

The second stage coils can be defrosted more quickly and more easily than the coils in the old bunker room.

Part of the load is carried at high suction pressure.

Since both evaporators are float controlled, less attention is required from the operator.

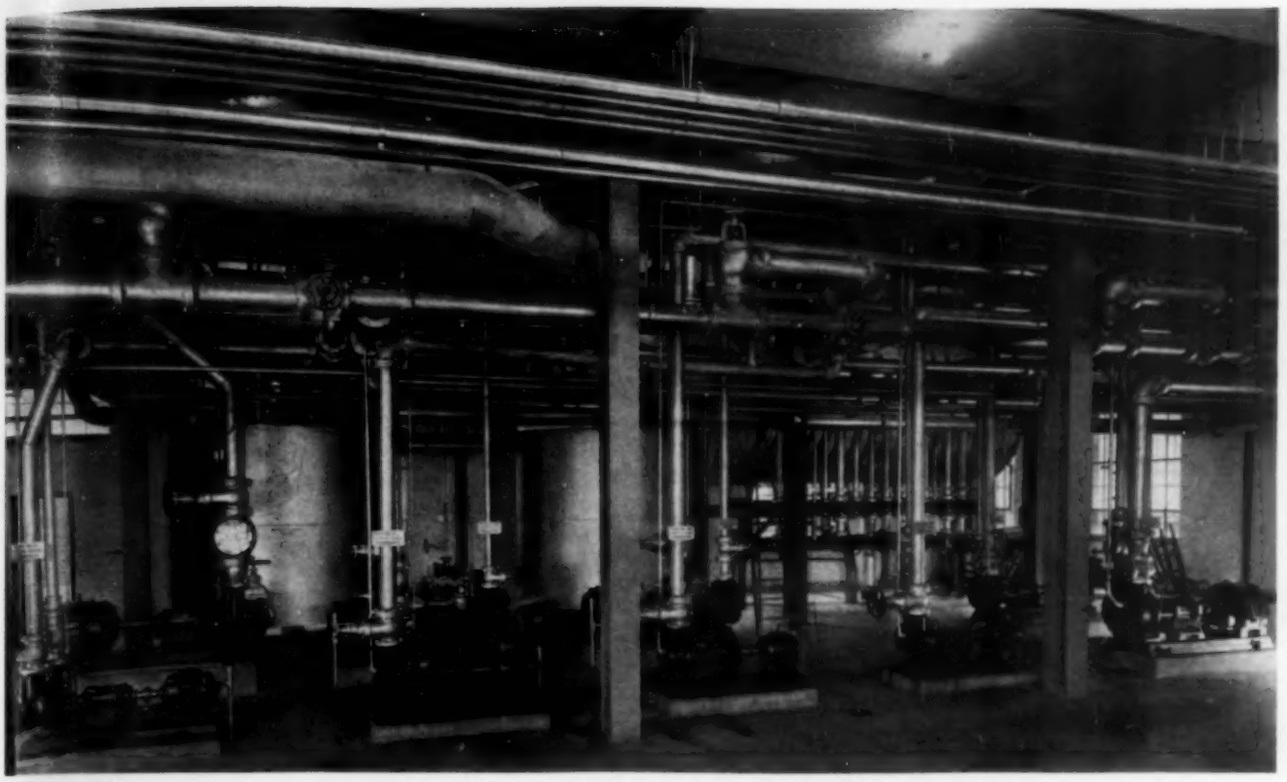
*Research Engineer, and Assistant Director of Development, respectively, Carrier Corp.

†U. S. Department of the Interior.

**R & H Chemicals Dept., E. I. du Pont de Nemours & Co.

††Engineering Dept., York Ice Machinery Corp.

[These papers have been published or will be published in full in *Refrigerating Engineering*, the journal of the ASRE.]



Battery of thin juice pumps at Amalgamated Sugar's new beet sugar factory. Note the foam piping on the end of each pump. This pipe leads back into the suction line

Piping Services in a Sugar Factory

By A. J. Forbess*

DESIGN features of the Amalgamated Sugar Co.'s new beet sugar factory at Nyssa, Oregon, include the use of welded steel piping, individual traps for the white pan coils, and lubricated plug cocks in the suction and discharge of the juice pumps. It was possible to simplify the piping layout because individual motor drives are used for the pumps and other pieces of equipment; where pumps and other equipment are driven from a line shaft and must be located near that shaft, long runs of piping are often called for.

The piping services include lines for high pressure steam, low pressure steam, vapor, high pressure water, low pressure water, air, CO_2 gas, SO_2 gas, beet juice, muriatic acid, soda ash solution, and beet pulp.

For high pressure saturated steam (215 lb pressure) extra heavy flanged steel pipe was installed. For low pressure steam, standard steel pipe with welded joints is used. Some of the low pressure or vapor lines are thin wall steel pipe.

For high pressure water, up to 300 lb (in the boiler feed line, for instance) extra heavy steel pipe with flanged fittings is employed. For low pressure water lines, standard and thin wall steel pipe is used. In the case of the main water line, laid underground from the Snake River to the factory (about 1450 ft) thin wall soil proof piping is used. The joints in this line were welded, each weld being treated and wrapped before being covered with soil. The fire line is 6 in. in diameter, 12 gage, spiral welded steel pipe, asphalt dipped, wrapped with fabric and protected on the outside with a layer of kraft paper. It was put together with bolted couplings.

High pressure air lines (pressure approximately 100 lb) are standard steel pipe. Low pressure air lines are 12 gage, spiral welded, undipped plain steel pipe.

The CO_2 gas line out of the lime kiln to the gas washer is cast iron. From the washer to the CO_2 compressor, steel pipe somewhat lighter than standard is employed.

For all beet juice lines, standard steel pipe is used. These lines vary in size from 4 to 10 in., depending on the quantity of juice to be handled, the viscosity and brix[1] of the juice, and on the amount of air and non-condensable gases present. The piping for muriatic acid is copper, and the piping for SO_2 gas from the sulphur station to the roof is asbestos-cement pipe. Standard steel pipe is used for soda ash solution.

For the underground sewer line from the factory to the Snake River, concrete sewer pipe was laid up to the point where the water from the pulp silo enters the sewer. From this point to the river vitrified tile was used, because of the acidity of the drainage from the pulp silo. The pulp line from the factory to the pulp silo, which is supported on posts 15 ft above the ground, is 10 in., 10 gage, spiral welded, undipped plain steel pipe.

Lubricated plug cocks are used in the suction and dis-

*Chief Engineer, Amalgamated Sugar Co.

[1]Brix denotes density or specific gravity of a sugar solution.

There is a trap for each coil of the white pans

charge of the juice pumps. This is an advantage because the plug cock will not scale up so that it cannot be closed; also, it will always hold the juice. It is often necessary to take the impeller out of a pump to remove scale. If the valves in the suction and discharge will not hold, removing the impeller becomes a very difficult and dangerous task as the juice may be very hot.

Each of the centrifugal pumps was designed for its special job. The suction and discharge are in most instances vertical, although where the pump is connected to a tank on the same floor, the suction is horizontal. Every juice pump is of a type that removes the air and non-condensable gases from the eye of the impeller. Because of this feature they will pump any amount of liquid, from very small amounts up to maximum capacity, without becoming vapor bound and without whipping or burning the juice.

Individual traps are used on the steam equipment, such as the white pan coils, thick and thin juice heaters, carbonation heaters, etc.

Battery of boiler feed pumps; there are four steam turbine and one motor driven units. Two pump condensate from the vacuum pan coils and one pumps condensate from the first effect evaporator calandria[2] directly to the boilers. Any make-up water required is pumped from the boiler feed tank, which is also condensed water and is at a temperature near the boiling point

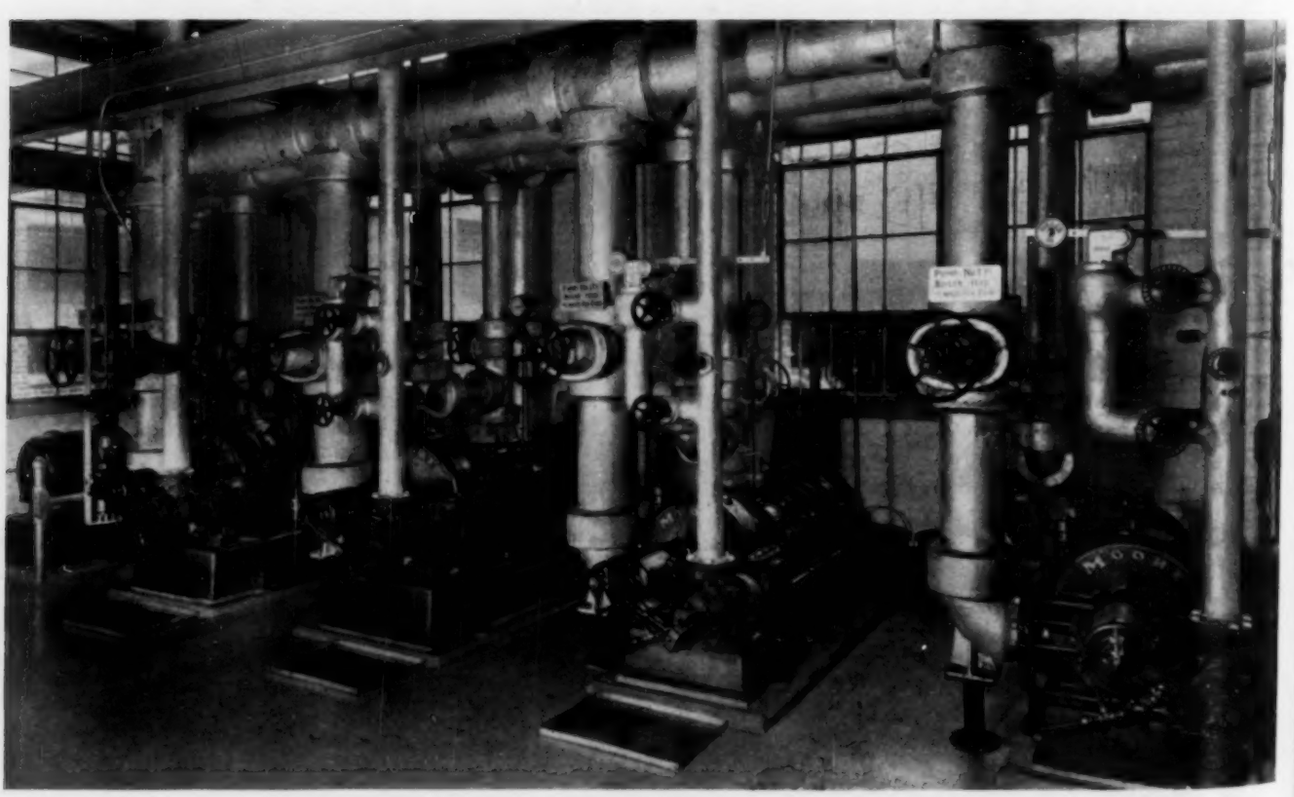
[2]The word "calandria," used in connection with vacuum pans and evaporators, means that the juice to be evaporated is on the inside of the coils and the steam is on the outside.

Hotel Installs Air Conditioning

SEVERAL hundred guest rooms in the Waldorf-Astoria hotel in New York City are being equipped with summer air conditioning. In addition to the guest rooms, approximately 350 tons of refrigeration will be installed to supply the cooling for several of the hotel's smaller ballrooms, private dining rooms, and a group of miscellaneous offices.

In the case of the guest rooms, conditioned air will be supplied to the room by a suspended air conditioning unit installed next to the ceiling of the room foyer, between the bathroom and clothes closet. A small grille outlet admits the cool air to the room proper. A subceiling under the unit completely conceals it and restores the original appearance and decoration. Other than the small circulating fan built as an integral part of the suspended air conditioning unit, no mechanism will be located in the guest room, as all other equipment is below the ground level. Cold water is the refrigerating medium, and is circulated to the suspended room units through a system of risers installed in a corner section of the clothes closets only a few inches from the unit locations. In a few cases, the ceiling of the closet itself is used as an installation point for the conditioning unit, with a grille opening into the room from that point.

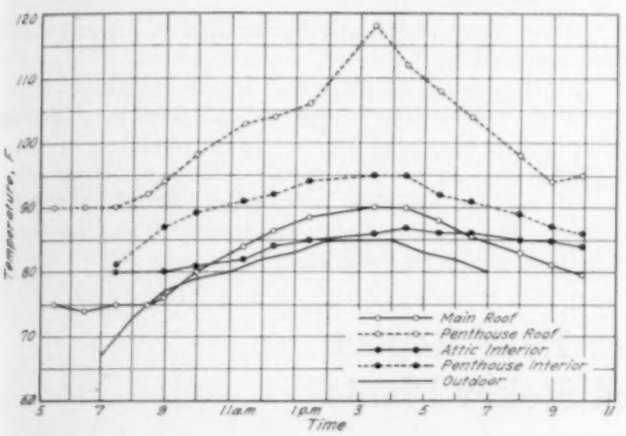
[*Concluded on p.* 446]



Temperature Study Shows Value of Flooding Roof to Reduce Heat Gain

By **R. A. Rothschild***

WHILE the Alabama highway department office building in Montgomery was under construction some two years ago, it became obvious that a high quality, very dense concrete was being secured throughout. This led to the idea that perhaps the built-up roof called for on the plans could be omitted. The subroof is a flat, reinforced concrete deck of about 15,000 sq ft area. With no covering over it the chief difficulty would obviously be due to temperature changes. It was therefore decided to keep the roof flooded with water; with this in mind, the entire roof was cast in one continuous pour with no joints whatever, and tied rigidly to the parapet wall surrounding. Since pouring, it has been kept covered with from 4 to 6 in. of water.



Temperature readings were taken on a warm, sunny day for the flooded main roof, the unprotected penthouse roof, the main attic interior, and the penthouse interior. The reported outdoor temperatures are also plotted here

Naturally, in an area as large as 15,000 sq ft, some shrinkage cracks developed immediately. Some of these were painted over with a narrow band of asphalt, while others closed of their own accord and appear to be healing autogenously.

It has been obvious from the first to those who have kept the roof under close observation that the water has proved an excellent insulator. Probably no feature of the building has caused so much comment as this roof, both from the general public and the engineering profession. While sprayed and even ponded roofs are not uncommon, this is the first example to come to our attention of water standing on an unprotected concrete deck.

While general success of the roof was plain, no quantitative statement of its value could be made, so it was recently determined to run a series of tests to obtain more definite information. For comparison with the test, something in the nature of a test specimen was needed; for this purpose, the roof of the elevator machine room was used. This machine room rises, as a penthouse, from the middle of the main roof. It consists of a concrete slab similar in construction to the roof slab except that it is much smaller and unrestrained at the edges. It has no covering, being exposed directly to the weather.

Immediately under and touching each of the roof slabs thermometers were placed. The thermometers and about 1 sq ft of the adjacent slab were covered with an inch of rock wool insulation, wedged tightly against the ceiling by a prop from the floor. Provision was made for withdrawing the instruments and taking rapid readings of them before they could be affected by the room temperature.

By largely preventing heat flow to or from the inside surface of the roof, thus stopping any heat flow through the slab, this arrangement should cause the top and bottom of the slab to approach the same temperature. It is felt that the thermometer readings accurately reflect the temperature conditions somewhere near the top (that is, the hottest) surface of the slab.

These thermometers were supplemented by two others, one at about the breathing line in the penthouse and the other at a similar height in the main attic—that is, the space under the main roof.

Test Results Plotted

The tests were run on May 4, which was a warm, sunny day. Readings were taken with some irregularity, but approximately hourly from 5:30 a.m. to 10:00 p.m.; these readings are plotted on the accompanying chart. The hourly temperatures published by the local weather bureau are also plotted on the chart.

From the chart it is at once apparent how the water insulation flattened out the temperature curves. The total ranges in temperature during the period of observation were: penthouse slab, 28 deg; outdoor temperature, 18; main roof slab, 16; machine room interior, 14; and attic interior, 7 deg.

In drawing conclusions from these figures, however,

Probably no feature of the Alabama highway department's building at Montgomery has caused as much comment as its flooded roof; while sprayed or ponded roofs are not uncommon, this is the first example in the author's knowledge of water standing on an unprotected concrete deck. . . . In order to see definitely the effect of the water as an insulator, some temperature studies were made which are reported

*Engineer, Alabama Highway Dept.

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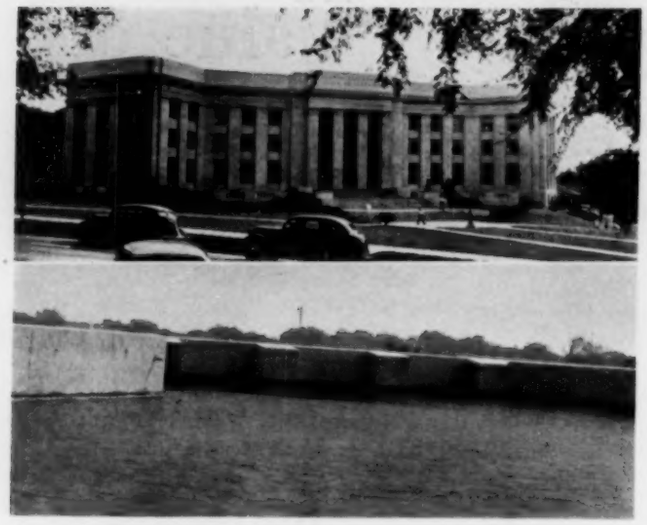


The Alabama highway department building, above, and a view of the water standing on the unprotected concrete roof deck

the interior temperature of the machine room cannot be relied upon too greatly as the readings were subject to two sources of error, (1) the heating effect of the elevator machinery, probably small; and (2) the effect of windows. While the attic space is entirely enclosed, with neither glass area nor ventilation to consider, the penthouse has a number of windows which allow more direct solar radiation and also allow ventilation. As a matter of fact, the windows were opened shortly before the peak temperature was reached, possibly flattening out the curve somewhat.

A more reliable and interesting observation is the comparison of peak temperatures of the slabs, as follows: outdoor peak temperature, 85 deg; main roof peak temperature, 90; and penthouse roof peak temperature, 118 deg. The high slab temperatures are of course due to the direct rays of the sun. Apparently the water eliminated about 80 per cent of this effect.

The money value of the flooded roof is probably small in winter, but when air conditioning is installed it will probably be quite large. Provision for air conditioning was made in the original construction by placing refrigeration coils in the heating ducts. The compressed air control system is also designed to care for summer cooling operation, and space has been provided for compressors, condensers, etc. Refrigeration will therefore require no change in the present heating system or duct work, but merely the addition of the proper equipment. It seems likely that it will be installed as soon as funds are available.

We hope in the future to take further similar readings, both on hotter days, which are plentiful in Alabama, and on cold days next winter. In the meantime, without generalizing as to the overall efficiency of our roof pond, we can say that it is quite efficient in reflecting the sun's rays, and very satisfactory in keeping down the temperature of the attic.

Smoke Preventers Hear Report on Field Survey of Stoker Plants

FURNACES are not meant to be smoke consumers, according to research engineers of Battelle Memorial Institute who addressed the Smoke Prevention Association meeting last month in Milwaukee. The best way to keep chimneys free from smoke, they said, is to prevent smoke formation in the fuel bed, and underfeed stokers, because they are designed to do this, are essentially smokeless in operation. The report also suggested that smoke ordinances of some cities are needlessly restrictive in demanding certain furnace volumes and often require useless expense to obtain them.

Several hundred fuel engineers heard the report of an extensive field study of stokers in actual operation. This was made last winter in Columbus, St. Louis, Cincinnati, and Chicago by research engineers of the institute. Sponsors of the work were the Stoker Manufacturers' Association, the Institute of Boiler and Radiator Manufacturers, the Steel Heating Boiler Institute and the National Coal Association. Two members of the Smoke Prevention Association served on the steering committee that directed the work.

The purpose of the survey was to find how freedom from smoke is related to size and height of the firebox, for which minimum figures are specified by some city ordinances. The survey engineers, R. C. Cross and H. N. Ostborg, employed specially designed portable test equipment in which a photoelectric cell, or "electric eye," gave a continuous record of the smoke going up the stack. Flue gas analyses and rates of coal feed were also determined on the job, and the supposedly critical combustion space and setting height were likewise measured.

The stokers studied were in homes, apartment buildings, laundries, and small industrial plants. More than 100 were investigated, and 22 were subjected to a full 24 hr continuous test. Only about one out of eight was found to be producing smoke sufficient to make a possible violation of smoke ordinances, and no such violation, it is stated, could be traced to small combustion space or low setting.

Coal fired by stokers, according to the Battelle report, is likely to cause smoke only when manual attention is given to the fire or when the stoker shuts off, and no amount of combustion space could eliminate smoke caused in these ways. The elimination of the tendency for stokers to smoke when shutting down to await the renewed demand for heat was called a distinct challenge to the ingenuity of the stoker manufacturer.

Ralph A. Sherman, supervising fuel engineer at Battelle and co-author with Messrs. Cross and Ostborg of the report, recommended further research, and said that many other factors than combustion space and setting height would have to be studied under controlled conditions. Among those which appear to be equally or more important are the adjustment of the air supply, the provision of excess air for shut down periods, the effect of various types of coal, and the equally variable types of service for which stoker fired heating plants are installed.

Suggested Dimensions for Pipe Flanges and Flanged Fittings for 4000 Lb, 900F

By A. M. Houser*

THERE is need for a pipe flange and flanged fitting standard in a pressure and temperature class higher than 2500 lb for oil refinery service and for steam service. A 4000 lb pressure class has been set up in answer to this need which affords an appreciable economy in weight over the 2500 lb American Standard (which is generally considered exceptionally heavy). This economy is the result of (1) the elimination of screwed and lapped flanges, (2) better proportioning of flanges and bolts and, (3) use of alloy steel. In order to keep the flange size down to a minimum only the welding neck flange (now a recognized design) is set up as the standard companion flange for the 4000 lb class.

Table 1—4 to 6 Per Cent Chromium Steel C5, ASTM Specification A157, Revised

Tensile strength, lb per sq in. (min)	100,000
Yield point, lb per sq in. (min)	65,000
Elongation in 2 in., per cent (min)	18
Reduction in area, per cent (min)	30
Carbon, per cent	0.15 to 0.35
Chromium, per cent	4.00 to 6.50
Molybdenum, per cent	0.40 to 0.65
Silicon, per cent	0.20 to 1.50
Manganese, per cent	1.00 max
Phosphorus, per cent	0.05 max
Sulphur, per cent	0.06 max

This provides 4000 lb flanges with diameters reasonably close to those of the 1500 lb flanges with thicknesses nearing those of the 2500 lb flange.

Materials and Ratings—The proposed dimensions are based on 4-6 per cent chromium steel for 4000 lb at 900 F. The requirements for the alloy steel used in this standard are taken from ASTM A-157, revised. (See Table 1.)

Type of Joint—The dimensions for the 4000 lb flanges are based on the ring joint, and this type of joint is primarily recommended for the 4000 lb standard. However, other equivalent types of joints can be used if desired.

Ring Joint Gaskets—The gasket dimensions are based upon a stress on the sectional area of the gasket, in the plane of the gasket, not exceeding 30,000 lb per sq in., being proportioned with relation to the bolting area in keeping with ring joint gaskets for the lower pressures. The ratio of the total gasket area to the total net bolt area for each size is as follows:

Size	Ratio	Size	Ratio
1 in.	1.84	2½ in.	2.09
1½ in.	2.05	3 in.	1.75
2 in.	2.03	4 in.	1.55

The pitch diameter of the ring joint gasket is fixed as small as possible with due regard to clearance between the ring groove and the inside diameter of the fitting.

Bolting—The load carried by each bolt stud when tightened is estimated for the probable unit stress on the net or basic minor diameter of each bolt stud based on the general practice of pulling up these bolts with a wrench. Table 2 gives these data for convenient reference. The relationships between the total bolt load and the total internal pressure are also tabulated and show a ratio of approximately four.

Flange Thicknesses and Stresses—Flange thicknesses have been so selected that the computed stress in flanges and blind flanges will not exceed 17,500 lb and those in the hubs will not exceed 25,000 lb per sq in. The external loads used in determining these stresses are the estimated installation bolt loads which will be applied in service.

Table 2—Bolting Data

Fitting Size	Bolting		Basic Minor Diam.	Basic Minor Area	Unit Bolt Stress	Load on One Bolt	Total Bolt Load W	Total Press Load (Ring Gasket) 4000 Lb per Sq In.	Ratio
	No.	Diam.							
1	4	⅞	0.7307	0.4193	48110	20175	80700	15900	5.07
1½	4	⅞	0.7307	0.4193	48110	20175	80700	19640	4.11
2	8	¾	0.6201	0.3020	51960	15700	125600	30680	4.09
2½	8	⅞	0.7307	0.4193	48110	20175	161400	39870	4.05
3	8	1	0.8376	0.5510	45000	24800	198400	48710	4.07
4	8	1¼	1.0876	0.9290	40250	37400	299200	74660	4.01

Wall Thicknesses—The minimum metal thicknesses recorded in Table 3 are one and one-half times the wall thickness which, according to Lamé's formula, will be stressed to 10,000 lb per sq in. by an internal pressure of 4000 lb per sq in.

Sizes and Dimensions for 4000 lb, 900 F—When preparing the 4000 lb standard it was intended to include sizes up to 12 in., but the further revisions required when the final draft was prepared were made only on sizes up to 4 in. Therefore, sizes above 4 in. need further study and are not included in this article. The dimension tables (Tables 3 and 4) covering flanges, drilling templates and the ring joints also include center to flange edge for ells, tees and crosses, and as the dimensions for valves for this pressure class must be greater than fittings, they also are given. In Table 3, dimension *J* is covered by the usual note, "to be specified by the purchaser." The dimension *S* is given in two columns, one for steam and one for oil, with a footnote directing the purchaser to the fact that the inside diameters of the fittings differ depending on the service. For steam or where corrosion is not a factor the port opening as given is for the size that the calculations were based on. For oil service where corrosion is a factor the inside diameter of the fitting is reduced. This is due to the fact that the flange drilling is as close as practical on the basis of allowance for casting on the outside only. The fittings are well proportioned even in the smaller sizes where in all classes or weights they seem a little out of

*Engineer of Standardization, Crane Co. Material reproduced with permission from a series of articles on standardization of piping equipment published in the *Valve World*.

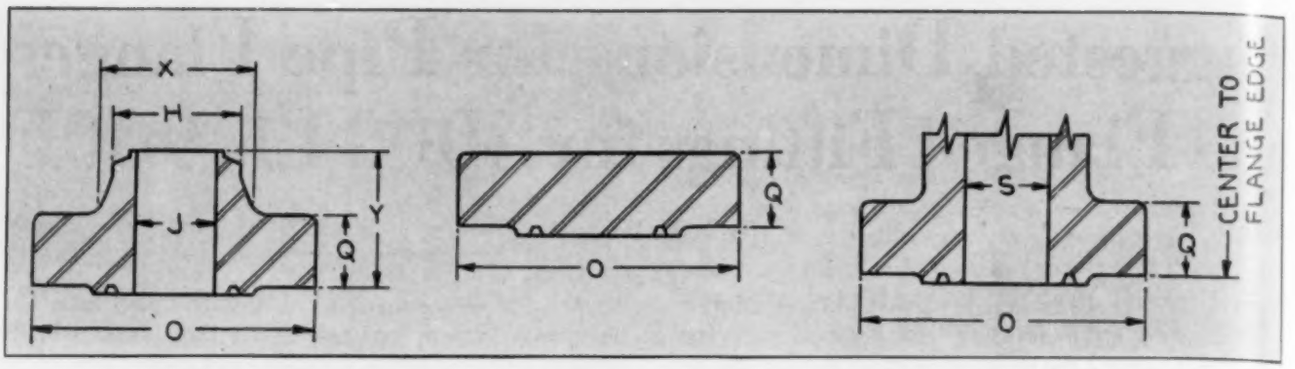


Table 3—Dimensions of Steel Flanges and Flanged Fittings for 4000 Lb Per Sq In (Gage)

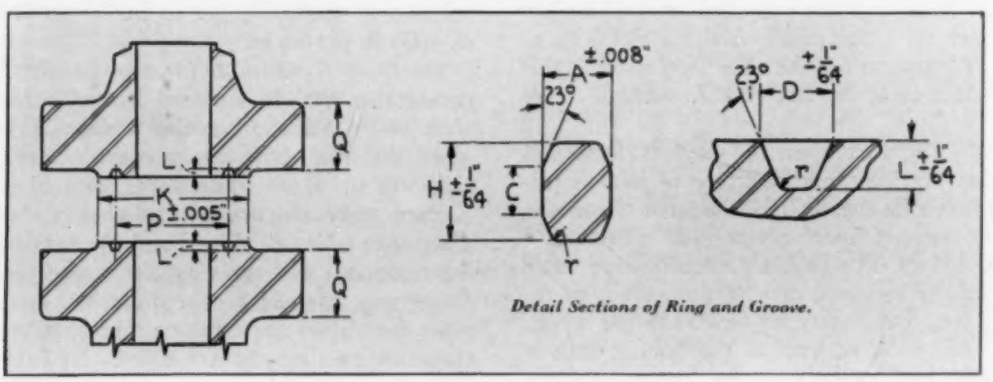
Nominal Pipe Size	Diam. of Flange	Thickness of Flange Min[c]	Diam. of Hub	Hub Diam. at Beginning of Chamfer	Length Through Hub[c]	Inside Diam of Pipe[b]	Diam. of Bolt Circle	No. of Bolts	Size of Bolts	Steam[a] Service		Oil[a] Service		Center to Flange Edge[c]	
										Inside Diam.	Metal Thickness Min	Corrosion Allowance	Inside Diam.	Ells, Tees and Crosses	Globe Valve
	O	Q	X	H	Y	J				S			S		
1	6½	1 11/16	1 7/8	1.32	3 3/16	To be specified by purchaser	4½	4	7/8	3/4	½	1/8	½	5 3/4	6 1/4
1½	7	1¾	2 3/8	1.90	3 3/4		5	4	7/8	1 1/4	½	1/8	1	6½	7 1/4
2	7½	2	3 1/16	2.38	4 3/16		5 3/4	8	3/4	1 5/8	11/16	1/8	1 3/16	7 1/4	...
2½	8 5/8	2 3/16	3 3/4	2.88	4 5/8		6 5/8	8	7/8	2	13/16	3/16	1 5/8	8 1/8	10
3	9 3/4	2 5/8	4 5/16	3.50	5 1/4		7½	8	1	2 3/16	15/16	3/16	2	9½	11
4	12¼	3 1/8	5 5/16	4.50	5 3/4		9½	8	1¼	3	1 3/16	3/16	2 5/8	11 3/4	...

[a]The inside diameter (*S*) depends on service conditions and should be specified by purchaser.
For steam service, no allowance is made for corrosion.
For oil service, if a corrosion allowance is required it must be taken on the inside of the fitting; therefore, the inside diameter for oil service is smaller than for steam service.
[b]The bore at the flange face for welding neck flanges should never be greater than column *S* for steam.
[c]The raised face for ring joint is not included in "thickness of flange, minimum," "length through hub," nor in "center to flange edge" dimensions for valves and fittings.

proportion, which, of course, is due to the size of nut and clearance needed regardless of the size.

The welding neck flange, being applied as regular practice, provides what now is considered good engineering, and in this case not only establishes the flange to pipe connection but provides the smallest diameter of bolts that can be used and maintain proper stress limits. For the higher pressures, and where welding neck flanges are adaptable, the use of this weight of fitting and flange should become common practice.

These dimensions were checked and discussed quite freely by a number of engineers qualified to pass judgment on this work and in no case were the flanges considered of insufficient strength, and the proportions were deemed practical from both a user and producer point of view.



Detail Sections of Ring and Groove.

Table 4—Facing Dimensions for 4000 Lb Flanges

Nominal Pipe Size	Pitch Diam. of Ring and Groove	Width of Ring	Height of Ring	Height of Flat on Ring	Width of Groove	Depth of Groove[a]	Diam. of Raised Face for Ring Jt.	Approx. Dis. Between Flgs. of Ring Jt. When Ring[b] Compressed	Fillet Radius
	P	A	H	C	D	L	K		r
1	2¼	7/16	5/16	1/16	15/32	5/16	3½	5/16	1/16
1½	2½	7/16	5/16	1/16	15/32	5/16	3 3/4	5/16	1/16
2	3 1/16	½	11/32	1/16	17/32	5/16	4 7/16	5/16	1/16
2½	3 5/8	5/8	13/32	1/16	21/32	7/16	5 1/16	5/16	1/16
3	3 15/16	5/8	13/32	1/16	21/32	7/16	5 3/4	5/16	1/16
4	4 15/16	3/4	15/32	1/16	25/32	½	7¼	5/16	1/16

Dimension *Q* is minimum flange thickness.
[a]Depth of groove is added to minimum flange thickness.
[b]For calculating "laying length" of fittings with ring joints the space dimensions given in the table must be added.

Properties of Mixtures of Air and Saturated Water Vapor for Barometric Pressures from 22 In. to 32 In. of Mercury

By William Goodman*

Barometric Pressure 22 In. of Mercury

All tabulated values (except temperatures and pressures) are quantities per pound of dry air in the mixture.

Temp. Deg Fahr	Absolute Humidity		Volume				Enthalpy (Heat Content)		Temp. Deg Fahr	Absolute Humidity		Volume				Enthalpy (Heat Content)	
	Pounds	Grains	Dry Air Cu Ft	Volume Increase of Humid Air Per Cent	Humid Specific Heat	1061*w*	Water Vapor Btu	Air-Vapor Mixture Btu		Pounds	Grains	Dry Air Cu Ft	Volume Increase of Humid Air Per Cent	Humid Specific Heat	1061*w*	Water Vapor Btu	Air-Vapor Mixture Btu
t	*w*	*w*	*v*		*s*		h_v	*h*	*t*	*w*	*w*	*v*		*s*		h_v	*h*
1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
−40	0.0001580	1.106	14.39	0.02540	0.2401	0.1676	0.1648	−9.435	50	0.01043	73.02	17.47	1.676	0.2447	11.07	11.30	23.30
−39	.0001676	1.173	14.42	.02694	.2401	.1778	.1748	−9.185	51	.01084	75.85	17.51	1.741	.2449	11.50	11.75	23.99
−38	.0001775	1.243	14.46	.02854	.2401	.1884	.1853	−8.935	52	.01125	78.72	17.54	1.808	.2451	11.93	12.20	24.68
−37	.0001880	1.316	14.49	.03022	.2401	.1994	.1963	−8.684	53	.01168	81.78	17.58	1.877	.2453	12.40	12.68	25.40
−36	.0001989	1.393	14.53	.03198	.2401	.2111	.2078	−8.432	54	.01212	84.85	17.61	1.948	.2455	12.86	13.16	26.12
−35	0.0002104	1.473	14.56	0.03382	0.2401	0.2232	0.2199	−8.180	55	0.01258	88.07	17.64	2.022	0.2457	13.35	13.66	26.86
−34	.0002226	1.558	14.59	.03579	.2401	.2362	.2328	−7.927	56	.01306	91.41	17.68	2.098	.2459	13.86	14.19	27.63
−33	.0002354	1.648	14.63	.03784	.2401	.2497	.2462	−7.674	57	.01355	94.83	17.71	2.177	.2461	14.37	14.72	28.40
−32	.0002489	1.742	14.66	.04002	.2401	.2641	.2605	−7.420	58	.01406	98.42	17.75	2.259	.2463	14.92	15.29	29.21
−31	.0002628	1.840	14.70	.04225	.2401	.2789	.2752	−7.165	59	.01458	102.1	17.78	2.342	.2466	15.47	15.86	30.02
−30	0.0002779	1.945	14.73	0.04468	0.2401	0.2948	0.2911	−6.909	60	0.01512	105.8	17.82	2.429	0.2468	16.04	16.45	30.85
−29	.0002936	2.055	14.76	.04720	.2401	.3115	.3077	−6.652	61	.01568	109.8	17.85	2.520	.2471	16.64	17.07	31.71
−28	.0003097	2.168	14.80	.04980	.2401	.3286	.3247	−6.395	62	.01627	113.9	17.88	2.612	.2473	17.26	17.72	32.60
−27	.0003270	2.289	14.83	.05257	.2402	.3470	.3430	−6.137	63	.01687	118.1	17.92	2.709	.2476	17.90	18.37	33.49
−26	.0003448	2.414	14.87	.05544	.2402	.3659	.3618	−5.878	64	.01748	122.3	17.95	2.808	.2479	18.54	19.05	34.41
−25	0.0003638	2.547	14.90	0.05849	0.2402	0.3860	0.3819	−5.618	65	0.01813	126.9	17.99	2.910	0.2482	19.23	19.76	35.36
−24	.0003839	2.687	14.94	.06172	.2402	.4073	.4032	−5.357	66	.01878	131.5	18.02	3.016	.2485	19.92	20.48	36.32
−23	.0004048	2.834	14.97	.06509	.2402	.4295	.4254	−5.095	67	.01946	136.2	18.06	3.125	.2488	20.65	21.23	37.31
−22	.0004267	2.987	15.00	.06859	.2402	.4527	.4485	−4.832	68	.02017	141.2	18.09	3.239	.2491	21.40	22.01	38.33
−21	.0004493	3.145	15.04	.07224	.2402	.4767	.4725	−4.568	69	.02090	146.3	18.12	3.356	.2494	22.17	22.82	39.38
−20	0.0004736	3.316	15.07	0.07615	0.2402	0.5025	0.4983	−4.302	70	0.02165	151.6	18.16	3.477	0.2497	22.97	23.65	40.45
−19	.0004989	3.492	15.11	.08020	.2402	.5293	.5251	−4.035	71	.02242	157.0	18.19	3.601	.2501	23.79	24.50	41.54
−18	.0005252	3.676	15.14	.08444	.2402	.5572	.5530	−3.767	72	.02323	162.6	18.23	3.730	.2505	24.65	25.40	42.68
−17	.0005527	3.869	15.18	.08885	.2403	.5864	.5822	−3.498	73	.02406	168.4	18.26	3.864	.2508	25.53	26.31	43.84
−16	.0005818	4.073	15.21	.09354	.2403	.6173	.6131	−3.227	74	.02492	174.4	18.30	4.001	.2512	26.44	27.26	45.02
−15	0.0006119	4.283	15.24	0.09837	0.2403	0.6492	0.6451	−2.955	75	0.02579	180.5	18.33	4.141	0.2516	27.36	28.23	46.24
−14	.0006436	4.505	15.28	.1035	.2403	.6828	.6788	−2.681	76	.02670	186.9	18.36	4.287	.2520	28.33	29.24	47.48
−13	.0006770	4.739	15.31	.1089	.2403	.7183	.7144	−2.406	77	.02766	193.6	18.40	4.441	.2525	29.35	30.30	48.78
−12	.0007119	4.983	15.35	.1145	.2403	.7553	.7514	−2.129	78	.02863	200.4	18.43	4.596	.2529	30.37	31.37	50.09
−11	.0007476	5.233	15.38	.1202	.2403	.7932	.7895	−1.851	79	.02963	207.4	18.47	4.757	.2533	31.43	32.48	51.44
−10	0.0007861	5.503	15.42	0.1264	0.2404	0.8341	0.8305	−1.570	80	0.03066	214.6	18.50	4.922	0.2538	32.53	33.62	52.82
−9	.0008256	5.779	15.45	.1327	.2404	0.8759	0.8726	−1.287	81	.03174	222.2	18.54	5.095	.2543	33.67	34.82	54.26
−8	.0008675	6.073	15.48	.1395	.2404	0.9204	0.9173	−1.003	82	.03284	229.9	18.57	5.271	.2548	34.84	36.04	55.72
−7	.0009103	6.372	15.52	.1464	.2404	0.9658	0.9630	−0.7170	83	.03398	237.9	18.60	5.455	.2553	36.05	37.31	57.23
−6	.0009554	6.688	15.55	.1536	.2404	1.014	1.011	−0.4289	84	.03514	246.0	18.64	5.641	.2558	37.28	38.60	58.76
−5	0.001003	7.023	15.59	0.1613	0.2405	1.065	1.062	−0.1377	85	0.03636	254.5	18.67	5.836	0.2564	38.57	39.95	60.35
−4	.001052	7.367	15.62	.1692	.2405	1.117	1.115	+0.1545	86	.03762	263.3	18.71	6.037	.2569	39.91	41.35	61.99
−3	.001104	7.726	15.66	.1775	.2405	1.171	1.170	0.4496	87	.03891	272.4	18.74	6.244	.2575	41.28	42.79	63.67
−2	.001158	8.105	15.69	.1862	.2405	1.229	1.228	0.7475	88	.04023	281.6	18.78	6.457	.2581	42.69	44.26	65.38
−1	.001215	8.503	15.72	.1953	.2406	1.289	1.288	1.048	89	.04162	291.3	18.81	6.681	.2587	44.16	45.80	67.16
0	0.001272	8.906	15.76	0.2046	0.2406	1.350	1.350	1.350	90	0.04305	301.3	18.84	6.907	0.2594	45.67	47.39	68.99
1	.001333	9.332	15.79	.2143	.2406	1.414	1.415	1.655	91	.04452	311.6	18.88	7.144	.2600	47.24	49.03	70.87
2	.001397	9.777	15.83	.2246	.2406	1.482	1.483	1.963	92	.04602	322.1	18.91	7.385	.2607	48.82	50.70	72.78
3	.001463	10.24	15.86	.2352	.2407	1.554	1.555	2.275	93	.04759	333.1	18.95	7.636	.2614	50.49	52.45	74.77
4	.001533	10.73	15.90	.2465	.2407	1.627	1.630	2.590	94	.04921	344.5	18.98	7.895	.2621	52.21	54.26	76.82
5	0.001605	11.24	15.93	0.2580	0.2407	1.703	1.707	2.907	95	0.05087	356.1	19.02	8.161	0.2629	53.97	56.12	78.92
6	.001680	11.76	15.96	.2700	.2408	1.782	1.787	3.227	96	.05258	368.1	19.05	8.436	.2637	55.79	58.02	81.06
7	.001758	12.31	16.00	.2826	.2408	1.865	1.871	3.551	97	.05434	380.4	19.08	8.719	.2645	57.66	59.99	83.27
8	.001840	12.88	16.03	.2957	.2408	1.952	1.958	3.878	98	.05619	393.4	19.12	9.015	.2653	59.62	62.06	85.58
9	.001925	13.48	16.07	.3095	.2409	2.043	2.050	4.210	99	.05839	406.6	19.15	9.320	.2661	61.64	64.18	87.94
10	0.002013	14.09	16.10	0.3236	0.2409	2.136	2.145	4.545	100	0.06003	420.2	19.19	9.629	0.2670	63.69	66.34	90.34
11	.002106	14.75	16.14	.3386	.2410	2.235	2.245	4.885	101	.06204	434.3	19.22	9.953	.2679	65.83	68.60	92.84
12	.002202	15.41	16.17	.3540	.2410	2.336	2.348	5.228	102	.06413	448.9	19.26	10.29	.2689	68.04	70.93	95.41
13	.002299	16.10	16.20	.3697	.2410	2.440	2.453	5.573	103	.06629	464.0	19.29	10.63	.2698	70.33	73.34	98.06
14	.002406	16.84	16.24	.3868	.2411	2.552	2.568	5.928	104	.06851	479.5	19.32	10.99	.2708	72.69	75.83	100.79
15	0.002514	17.60	16.27	0.4041	0.2411	2.667	2.684	6.284	105	0.07078	495.5	19.36	11.35	0.2719	75.10	78.37	103.57
16	.002625	18.38	16.31	.4220	.2412	2.785	2.804	6.644	106	.07315	512.0	19.39	11.73	.2729	77.61	81.03	106.47
17	.002741	19.19	16.34	.4407	.2412	2.909	2.930	7.010	107	.07561	529.3	19.43	12.12	.2740	80.22	83.79	109.47
18	.002863	20.04	16.38	.4603	.2413	3.038	3.061	7.381	108	.07813	546.9	19.46	12.53	.2752	82.89	86.61	112.53
19	.002989	20.92	16.41	.4805	.2413	3.171	3.197	7.757	109	.08074	565.2	19.50	12.94	.2763	85.67	89.54	115.70
20	0.003120	21.84	16.44	0.5016	0.2414	3.310	3.338	8.138	110	0.08344	584.1	19.53	13.38	0.2776	88.53	92.58	118.98
21	.003257	22.80	16.48	.5236	.2415	3.456	3.487	8.527	111	.08620	603.4	19.56	13.82	.2788	91.46	95.68	122.32
22	.003397	23.78	16.51	.5461	.2415	3.604	3.638	8.918	112	.08909	623.6	19.60	14.28	.2801	94.53	98.92	125.80
23	.003543	24.80	16.55	.5696	.2416	3.759	3.796	9.316	113	.09208	644.5	19.63	14.76	.2814	97.69	102.27	129.39
24	.003697	25.88	16.58	.5944	.2417	3.922	3.962	9.722	114	.09515	666.0	19.67	15.25	.2828	100.95	105.72	133.08
25	0.003852	26.96	16.62	0.6193	0.2417	4.087	4.130	10.13	115	0.09832	688.2	19.70	15.75	0.2842	104.31	109.29	136.89
26	.004021	28.15	16.65	.6464	.2418	4.266	4.313	10.55	116	.1016	711.4	19.74	16.28	.2857	107.83	113.01	140.85
27	.004190	29.33	16.68	.6736	.2419	4.445	4.496	10.98	117	.1050	735.3	19.77	16.83	.2873	111.45	116.85	144.93
28	.004365	30.55	16.72	.7017	.2420	4.631	4.686	11.41	118	.1085	759.7	19.80	17.39	.2888	115.15	120.77	149.09
29	.004548	31.84	16.75	.7312	.2421	4.826	4.885	11.85	119	.1121	784.9	19.84	17.97	.2905	118.97	124.82	153.38
30	0.004740	33.18	16.79	0.7621	0.2421	5.030	5.094	12.29	120	0.1159	811.5	19.87	18.58	0.2922	123.00	129.11	157.91
31	.004941	34.59	16.82	.7944	.2422	5.243	5.312	12.75	121	.1198	838.7	19.91	19.19	.2939	127.13	133.49	162.53
32	.005141	35.99	16.86	.8263	.2423	5.454	5.531	13.21	122	.1238	866.6	19.94	19.84	.2957	131.35	137.98	167.26
33	.005358	37.50	16.89	.8611	.2424	5.685	5.766	13.69	123	.1280	895.9	19.98	20.50	.2976	135.80	142.70	172.22
34	.005579	39.06	16.92	.8968	.2425	5.920	6.007	14.17	124	.1323	926.2	20.01	21.20	.2995	140.38	147.57	177.33
35	0.005807	40.65	16.96	0.9335	0.2426	6.162	6.255	14.66	125	0.1368	957.5	20.04	21.91	0.3016	145.12	152.62	182.62
36	.006047	42.33	16.99	0.9720	.2427	6.416	6.517	15.16	126	.1414	989.9	20.08	22.65	.3036	150.05	157.85	188.09
37	.006295	44.07	17.03	1.012	.2428	6.679	6.786	15.67	127	.1462	1023.6	20.11	23.41	.3058	155.15	163.28	193.76
38	.006551	45.86	17.06	1.053	.2430	6.950	7.064	16.18	128	.1512	1058.5	20.15	24.22	.3081	160.44	168.91	199.63
39	.006815	47.71	17.10	1.095	.2431	7.231	7.353	16.71	129	.1564	1094.7	20.18	25.05	.3104	165.93	174.75	205.71
40	0.007090	49.63	17.13	1.139	0.2432	7.522	7.652	17.25	130	0.1618	1132.8	20.22	25.90	0.3128	171.70	180.91	212.11
41	.007373	51.61	17.16	1.185	.2433	7.822	7.960	17.80	131	.1674	1171.7	20.25	26.79	.3153	177.60	187.19	218.63
42	.007667	53.67	17.20	1.232	.2435	8.135	8.282	18.36	132	.1730	1211.1	20.28	27.70	.3179	183.56	193.55	225.23
43	.007970	55.79	17.23	1.281	.2436	8.456	8.613	18.93	133	.1790	1253.3	20.32	28.66	.3206	189.96	200.36	232.28
44	.008285	57.99	17.27	1.332	.2437	8.790	8.956	19.52	134	.1853	1297.0	20.35	29.65	.3234	196.59	207.43	239.59
45	0.008615	60.31	17.30	1.384	0.2439	9.141	9.317	20.12	135	0.1917	1342.0	20.39	30.69	0.3263	203.41	214.71	247.11
46	.008953	62.67	17.34	1.438	.2440	9.499	9.686	20.73	136	.1984	1388.7	20.42	31.75	.3293	210.49	222.26	254.90
47	.009301	65.11	17.37	1.495	.2442	9.869	10.07	21.35	137	.2054	1437.7	20.46	32.86	.3324	217.92	230.20	263.08
48	.009665	67.66	17.40	1.553	.2444	10.26	10.47	21.99	138	.2128	1489.4	20.49	34.04	.3358	225.75	238.56	271.68
49	.01004	70.28	17.44	1.613	.2445	10.65	10.88	22.64	139	.2203	1541.8	20.52	35.23	.3391	233.69	247.03	280.39

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Properties of Mixtures of Air and Saturated Water Vapor—(Continued)

Barometric Pressure 22 In. of Mercury

All tabulated values (except temperatures and pressures) are quantities per pound of dry air in the mixture.

Temp. Deg Fahr	Absolute Humidity		Volume		Humid Specific Heat	1061w	Enthalpy (Heat Content)	
	Pounds	Grains	Dry Air Cu Ft	Volume Increase of Humid Air Per Cent			Water Vapor Btu	Air-Vapor Mixture Btu
t	w	w	v		s		h_v	h
1	2	3	4	5	6	7	8	9
140	0.2281	1596.6	20.56	36.48	0.3426	242.00	255.92	289.52
141	.2362	1653.4	20.59	37.78	.3463	250.61	265.11	298.95
142	.2449	1714.2	20.63	39.15	.3502	259.83	274.96	309.04
143	.2537	1775.7	20.66	40.58	.3542	269.14	284.92	319.24
144	.2629	1840.4	20.70	42.05	.3583	278.95	295.41	329.97
145	0.2726	1908.2	20.73	43.60	0.3627	289.23	306.43	341.23
146	.2828	1979.9	20.76	45.22	.3673	300.09	318.05	353.09
147	.2934	2053.9	20.80	46.89	.3720	311.31	330.06	365.34
148	.3045	2131.4	20.83	48.66	.3770	323.05	342.63	378.15
149	.3160	2212.1	20.87	50.51	.3822	335.29	355.73	391.49
150	0.3283	2297.8	20.90	52.45	0.3877	348.28	369.65	405.65
151	.3410	2386.9	20.94	54.49	.3935	361.79	384.13	420.37
152	.3539	2477.4	20.97	56.60	.3993	375.51	398.83	435.31
153	.3684	2578.5	21.00	58.85	.4058	390.82	415.24	451.96
154	.3831	2681.4	21.04	61.18	.4124	406.42	431.97	468.93
155	0.3987	2790.7	21.07	63.67	0.4194	422.99	449.74	486.94
156	.4150	2904.7	21.11	66.27	.4267	440.27	468.32	505.76
157	.4320	3023.9	21.14	68.98	.4344	458.33	487.71	525.39
158	.4502	3151.4	21.18	71.88	.4426	477.66	508.46	546.38
159	.4693	3285.0	21.21	74.91	.4512	497.91	530.19	568.35
160	0.4896	3427.2	21.24	78.15	0.4603	519.47	553.35	591.75
161	.5110	3577.1	21.28	81.56	.4700	542.19	577.76	616.40
162	.5339	3737.3	21.31	85.20	.4803	566.47	603.84	642.72
163	.5580	3906.1	21.35	89.04	.4911	592.05	631.33	670.45
164	.5833	4087.0	21.38	93.16	.5027	619.48	660.81	700.17
165	0.6113	4278.8	21.42	97.51	0.5151	648.54	692.06	731.66
166	.6407	4485.1	21.45	102.2	.5283	679.81	725.69	765.53
167	.6720	4703.9	21.48	107.2	.5424	712.97	761.35	801.43
168	.7058	4940.9	21.52	112.6	.5576	748.90	800.00	840.32
169	.7418	5192.6	21.55	118.3	.5738	787.05	841.05	881.61
170	0.7807	5465.0	21.59	124.5	0.5913	828.34	885.49	926.29
171	.8229	5760.1	21.62	131.2	.6103	873.07	933.63	974.67
172	.8684	6079.1	21.66	138.4	.6308	921.41	985.68	1027.0
173	.9177	6423.8	21.69	146.3	.6530	973.67	1042.0	1083.5
174	.9716	6801.2	21.72	154.8	.6772	1030.9	1103.5	1145.3
175	1.030	7212.1	21.76	164.1	0.7036	1093.1	1170.6	1212.6
176	1.095	7665.7	21.79	174.4	.7328	1161.9	1244.7	1286.9
177	1.166	8163.4	21.83	185.8	.7648	1237.3	1326.0	1368.5
178	1.245	8716.4	21.86	198.3	.8003	1321.2	1416.3	1459.0
179	1.333	9331.7	21.90	212.3	.8399	1414.4	1516.7	1559.6
180	1.432	10.021	21.93	227.9	0.8842	1518.9	1629.3	1672.5
181	1.543	10.798	21.96	245.5	0.9341	1636.6	1756.1	1799.6
182	1.669	11.681	22.00	265.6	0.9909	1770.5	1900.5	1944.2
183	1.813	12.690	22.03	288.5	1.056	1923.5	2065.4	2109.4
184	1.980	13.861	22.07	315.0	1.131	2100.9	2256.7	2300.9
185	2.176	15.230	22.10	346.1	1.219	2308.4	2480.5	2524.9
186	2.406	16.844	22.14	382.8	1.323	2553.1	2744.4	2789.0
187	2.685	18.794	22.17	426.9	1.448	2848.7	3063.2	3108.1
188	3.026	21.181	22.20	481.1	1.602	3210.5	3453.5	3498.6
189	3.455	24.182	22.24	549.2	1.795	3665.2	3944.0	3989.4
190	4.006	28.043	22.27	636.8	2.043	4250.6	4575.1	4620.7
191	4.747	33.227	22.31	754.4	2.376	5036.2	5422.6	5468.5
192	5.769	40.380	22.34	919.9	2.836	6120.5	6592.4	6638.4
193	7.371	51.596	22.38	1170.9	3.557	7820.5	8426.4	8472.7
194	10.05	70.336	22.41	1596.2	4.762	10661	11491	11537
195	15.55	108.857	22.44	2470.0	7.238	16500	17790	17837
196	33.41	233.849	22.48	5305.4	15.27	35445	38231	38278
197			22.51					
198			22.55					
199			22.58					
200			22.62					

TABLES of the psychrometric properties of air for the normal barometric pressure of 29.921 in. have been used for a variety of pressures. However, there are many localities where the tables for normal pressure cannot be used because the average barometric pressure is considerably lower. In addition, there are times in experimental work where errors may result because the variation in barometric pressure is neglected. For these reasons, tables of the psychrometric properties of air have been prepared for barometric pressures from 22 in. to 32 in. of mercury in 1 in. intervals. One of these tables will be published each month. With the aid of these tables, accurate computations can be made under all conditions.

With a few exceptions, the tables are arranged and used in the same manner as the table for the normal barometric pressure of 29.921 in. of mercury (14.696 lb per sq in.) published in HEATING, PIPING AND AIR CONDITIONING, January, 1938. The columns giving the vapor pressure of moisture and the sensible heat of air are omitted in the new tables because these values are the same as those given in the table for normal barometric pressure. In addition, one column that did not appear in the table for normal pressure appears in the new tables. This is column 7 and is headed *1061w*. The values in this table are of use for quickly computing the *true* enthalpy (total heat) of an air-vapor mixture. For a further discussion of this point, see the January, 1938, HPAC. The quickest way of computing the true enthalpy of an air-vapor mixture is to use formula 3*b* of the January, 1938, article. This formula is

$$h = s''t + 1061w$$

where s'' is the humid specific heat of a saturated mixture at the dew point temperature; w is the absolute humidity; t is the dry bulb temperature; and h is the true enthalpy in Btu per lb of dry air in the mixture.

By means of this formula, the true enthalpy of an air-vapor mixture can be computed quickly if its dry bulb and dew point temperatures are known.

Example—Find the true enthalpy of humid air for a dry bulb temperature of 95 deg and a dew point temperature of 63 deg. The barometric pressure is 22 in.

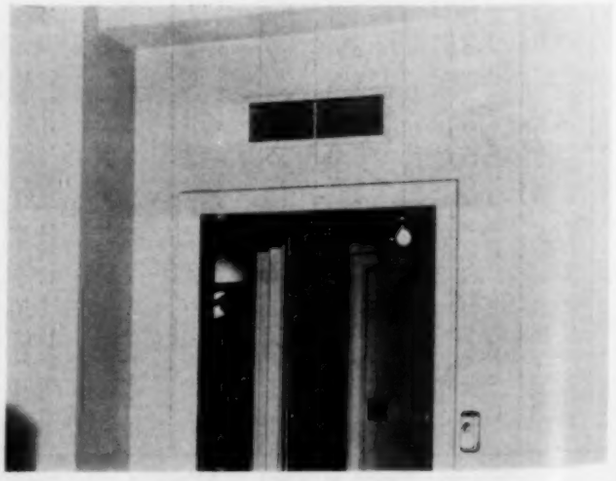
Solution—Referring to the table, at a dew point temperature of 63 deg, s'' = 0.2476 and 1061w = 17.90. Therefore, $h = 0.2476 \times 95 + 17.90 = 41.42$ Btu per lb.

[*Concluded from p.* 440]

Although the units used have been designed for quiet operation at full capacity, a two speed switch above the light switch on the wall provides a low speed that is suited to all-night operation. A thermostat mounted on the wall with the two speed switch allows the guest to adjust the temperature.—FRANK C. LYONS.*

*Frigidaire Div., General Motors Sales Corp.

Units are installed in the ceiling of the room foyer, and chilled water is circulated to the units through a riser system. View shows the grille over the door



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AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

Journal Section

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BULLETIN BOARD

Announcement of the appointment of Harold T. Olson to the staff of the ASHVE Research Laboratory, Pittsburgh, effective June 1, has been made by Chairman W. L. Fleisher, Committee on Research. Mr. Olson is a graduate of the University of Minnesota, BS 1933, MS 1936.

* * *

A two-year program of cooperative research with the United States Navy will be carried out under the auspices of the ASHVE Committee on Research and the studies will involve problems relating to the improvement of ventilation, reduction of noise, and increasing the efficiency of Uncle Sam's fighting ships and personnel.

* * *

On July 1 the total membership of the Society was 3127, composed of 2 Honorary Members, 64 Life Members, 1684 Members, 867 Associate Members, 435 Junior Members and 75 Student Members.

* * *

Members entitled to Volumes 43 and 44 of TRANSACTIONS will receive them about August 1, as production of these two volumes is now nearing completion.

* * *

THE 46TH ANNUAL MEETING, JANUARY 22-26, 1940, CLEVELAND, OHIO.

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Preview of Semi-Annual Meeting Papers*

Heating Low-Cost Homes

By Robert K. Thulman and L. E. Seeley

The response of people to thermal environment runs in a predictable manner and as standards of comfort have been established by the Society a challenge to the heating industry is presented to furnish practical and economical equipment.

Public health is improved by the increased use of ample quantities of domestic hot water service and it is suggested that low-cost home developments should have as a minimum supply 30 to 50 gallons of hot water available per day. While this domestic hot water need not be supplied by the heating system there will be instances where the heating plant will supply this water and in these cases the performance of the combined equipment must be considered.

Climatic conditions vary in different sections of the country making differences in heating costs seem inevitable and it is estimated that the maximum capacity of a heating system may vary as much as 400 per cent for a given structure. This range may be reduced by insulation, control of window sizes and generally better construction by about 100 per cent.

Low cost homes should not be analyzed in terms of first or initial costs but in terms of operating and maintenance costs. Present trends are towards development of individual heating units which can be installed regardless of whether there is a basement, and one feature of this design is that there is a tendency to confine equipment in a very small space.

Effect of Size and Type of Air Inlet and Outlet on the Heat Output of Convectors

By A. P. Kratz, M. K. Fahnestock and E. L. Broderick

Results of cooperative research between the University of Illinois and the Society are reported dealing with the effect of inlet and outlet design on the heat output of convectors. Data obtained from these tests indicate that the free area of a grilled inlet should be at least equal to, and of an ungrilled inlet should be approximately 25 per cent in excess of, the free area through the heating unit. For maximum heat output the free area of a convector outlet should not be less than 10 per cent nor greater than 30 per cent in excess of the free area through the unit.

Directing vanes installed at the inlet to a convector are not effective and may prove detrimental by introducing additional friction. It was found that the most effective position for a directing vane at the outlet is in a central location with respect to both the cabinet and the outlet face.

Heat outputs of convectors may be from 9.5 to 14.5 per cent greater with a top outlet than with a unit having a front outlet with the same grille free area. Convectors may be converted into humidifiers without loss in effectiveness as a heating device by installing a water pan provided with steam space in the top of the cabinet.

*To be published in later issues of ASHVE Journal Section.

Ozone in Ventilation—Its Possibilities and Limitations

By W. N. Witheridge and C. P. Yaglou

Does ozone counteract body odors? How are occupants affected by varying concentrations? Does ozone oxidize body odors in the air? Answers to these questions were studied in a series of observations conducted in experimental rooms at Harvard School of Public Health.

Results indicated that ozone in concentrations of 0.015 ppm, that can barely be smelled by the occupants of a room, reduced the smell of body odor sufficiently to permit a reduction of at least 50 per cent in the amount of outdoor air required for odor control. Higher concentrations were irritating to the mucous membranes of the upper respiratory tract, while lower concentrations had no effect on body odors. The action of ozone on body odors appears to take place not in the air of the room by oxidation as is sometimes assumed, but on the mucous membranes of the nasal passages by masking. Although ozone in sufficient strength completely obliterated the smell of body odors, it proved difficult to counteract the two so that neither could be smelled by persons entering the room from outdoor air. Ozone had to be present in perceptible concentrations to do any good at all.

The main objection to the application of ozone to ventilation is shown to be lack of control. Until adequate means are devised for controlling the output of ozonators from the actual concentration in occupied spaces, the use of ozone should be discouraged because of its great toxicity.

Cooling Tower Performance Studies

By L. M. K. Boelter

The progress report on cooperative research sponsored by the Society and conducted at the University of California reveals tentative results on the Evaporative Cooling Project dealing with drop dynamics which will be eventually augmented by the application of these data to the mass and heat transfer phenomena. Unique instruments designed and built for this study include a hair hygrometer for the determination of space distribution of humidity, a thermocouple psychrometer designed for free convection circulation and possessing a wick diameter of less than 1 mm, and a radiometer to measure the rate of energy radiated by the tower under operating conditions. Another instrument developed for this investigation is a pressure plate anemometer, having a natural frequency of moving parts sufficiently high to measure the velocity fluctuations which will be effective in the cooling of a water stream.

Models of the full-scale atmospheric towers placed in a wind stream of known turbulence permit studies of air flow through the equipment. Air-free water is being used to determine the effect of dissolved gases. Several ratios of cooling towers have been proposed based on the concept of an air heating device and of a water cooling device, all of which is important in making available design data of fundamental importance.

The Effect of Vibration Upon Free Convection from Horizontal Cylinders

By R. C. Martinelli and L. M. K. Boelter

This paper deals with a study of the rate of heat transfer from a 0.75 in. diameter horizontal cylinder, which is being vibrated sinusoidally in a vertical direction, to surrounding water. The amplitude of vibration of the cylinder was varied from zero to 0.10 in. and the frequency of vibration from zero to 40 cycles per second. The difference in temperature between the cylinder and the surrounding fluid was varied from 8 to 45 F.

The practical application of the results may be referred to the required amplitude of vibration which a ½ in. tube carrying 250 F steam and used to heat water to 150 F has to be moved in order to double the heat transfer rate. If the frequency of vibration is 30 cycles per second it follows from a computation based on the experimental data that the required tube amplitude would have to be 0.051 in.

If a body is losing heat in air, the vibration is not as effective in increasing the rate of heat transfer as it would be if water is the ambient fluid. It follows that vibrating a body in air will have no effect on the rate of heat transfer, until it is vibrated about 70 times as fast as would be necessary in water. For heat transfer to fluids similar to water, in apparatus subjected to vibration, a liberal design results if the apparatus behavior prediction is based upon data obtained from stationary equipment.

Fire Protection for Air Conditioning Systems

By R. C. Loughead

Air conditioning systems should be designed to eliminate potential fire hazards. Due to the inherent features of an air conditioning system it is impracticable to suggest specific fire protection that would properly apply to all cases and therefore it is necessary to give each system requiring protection special study and treatment. Most systems and especially ones which are confined to one floor or a part of a floor will usually not require any special fire protection if installed according to available regulations. However larger systems supplying more than one floor may present a hazard which should be protected regardless of installation methods. The best assurance against loss by fire is not only a sound engineering installation but the maintenance of cleanliness in the system at all times and constant vigilance by the owner or occupant.

Equipment is now being developed for the detection of smoke in air conditioning systems intended to reduce the possibilities of smoke damage and panic hazard. These devices are intended to shut down fans, close dampers, sound alarms and perform other necessary functions to prevent the spread of fire and smoke.

Although developments in the air conditioning industry are advancing rapidly many of the necessary regulations which have been established are in a continual state of flux. The industry is cooperating in the development of suitable protection as it has recognized that acceptable safety features must be incorporated in the design of these systems to eliminate the possibility of serious catastrophes.

Summer Cooling Requirements in Washington, D. C., and Other Metropolitan Districts

By F. C. Houghten, Carl Gutberlet and Albert A. Rosenberg

Comfort tests conducted at the Federal Reserve Building in Washington, D. C. confirm previous observations made in Minneapolis, San Antonio and in Toronto, Canada, to the effect that a practice of maintaining 71 deg effective temperature indoors with a tolerance of plus or minus 1 deg will satisfy practically all groups. Since 1935 the Research Laboratory of the Society has been studying the cooling requirements of persons seated at rest or engaged in sedentary occupations in summer cooled and air conditioned spaces. After a careful study of all the data it seems apparent there is a consistent desire for lower temperatures in Minneapolis and Toronto than in the other cities under observation.

Data collected in all these investigations apply to persons entering a cooled space and remaining for 3 hours or more, which may be construed as permanent occupancy. No serious shock was reported in any of the suggested effective temperatures to be maintained, and in those cases where shock was noted the experience would be the same whether entering the cooled space for 5 min or for permanent occupancy. Hence, the same conditions as maintained for permanent occupancy should give satisfactory results for occupancy of shorter duration.

Study of Summer Cooling in the Research Residence for the Summer of 1938

By A. P. Kratz, S. Konzo, M. K. Fahnestock, and E. L. Broderick

Where the initial cost of a residence cooling installation is more important than the maintenance of ideal indoor conditions, it was determined through tests conducted in 1938 at the Research Residence, University of Illinois, that a small capacity condensing unit was feasible. A cooling capacity of 18,000 Btu per hour was found to be sufficient for both stories of the residence on days where the maximum outdoor temperature rose to 95 F and the minimum outdoor temperature during the night preceding the test did not exceed 71 F.

Satisfactory cooling on both floors of the Research Residence was accomplished by the introduction of cooled air on only the second floor. However, the general application of this principle may be limited by the room arrangement on the first story and the location of the return air grilles as well as the location of rooms remote from the stairway.

Studies made with heat-flow meters indicated considerable discrepancy between measured and calculated values, and it was further noted that thermal equilibrium was never attained in a wall section under actual conditions. The application of insulation to the residence's frame wall construction reduced the heat flow rate, but the magnitude was much less than indicated by the ratio of the calculated coefficient of heat transmission for the insulated and uninsulated walls. The temperature of the inside wall surface was greatly reduced with the use of insulation which resulted in conditions more conducive to comfort for a given air temperature than those produced by an uninsulated wall.

A Theory Covering the Transfer of Vapor Through Materials

By **Frank B. Rowley*** (*MEMBER*), **Minneapolis, Minn.**

THERE has been much speculation about the theory relating to the transfer of vapor through materials and the application of the theory to building construction. For convenience it has often been assumed that the laws for vapor transmission are similar in form to those governing the flow of heat through the walls of a building, and that coefficients of vapor transmittance may be developed for materials or combinations of materials which may be applied in the same manner as coefficients of heat transmission. In making this analogy it is assumed that the difference in vapor pressures between two parts of a structure is the motive force which causes the flow of vapor and that the amount of vapor transmitted is directly proportional to the difference in vapor pressure and inversely proportional to the vapor resistance between the two parts of the wall. The over-all vapor resistance of a built-up wall section is assumed to be equal to the sum of the vapor resistances of its component parts. If the analogy were complete a vapor transmittance coefficient would be defined as the quantity of vapor transmitted per unit cross-sectional area, per unit time, per unit difference in vapor pressure along the path of transmittance, and there would be coefficients for surfaces, air spaces, homogeneous materials, and combinations of materials.

Before accepting a complete analogy between the two problems an analysis should be made to determine those elements which are similar and those which may be conflicting. Heat is a form of energy having no physical properties, and unless it is changed from sensible to latent, or vice versa, it will be transferred through a wall without change of state. Vapor is a substance with physical properties and in the course of its transfer through a structure may change its state several times. Heat may be transmitted by radiation, conduction, and convection. Vapor may be transmitted by molecular diffusion and convection, and the condensed vapors may be transmitted by capillarity or other means.

Water vapor in the air is steam and has the same physical properties that it would have if the air were not present. If air at a given temperature is saturated with water vapor, the density of the vapor in the air is the same as the density of saturated steam at the given temperature, and the pressure of the vapor in the air corresponds to the boiling pressure of water at the given temperature. If the temperature is raised the vapor is superheated, but the vapor pressure is not changed. If the temperature is lowered, some of the vapor is condensed and the vapor pressure and density are reduced to values corresponding to saturation at the new temperature.

The method by which water vapor travels from one point to another depends partly upon the material through which it travels and partly upon the temperature along its path. Insofar as its transfer through building construction is concerned the materials through which it travels may be classified as air and solids. The conditions of air which seem to be of greatest importance are temperature, pressure, and air movement. The properties of the solids which seem to have the greatest significance are permeability to gases and power to absorb and condense water vapor from the air. These two properties of solid materials together with temperature largely determine the state of the moisture as it travels through them.

If the temperature of air is above the dew-point temperature of vapor the vapor may be transmitted through the air by turbulence and by molecular diffusion. The transfer of vapor by convection currents is similar to the transfer of heat by convection currents. The resistance to the flow of vapor through air by molecular diffusion is a function of density and temperature of the air through which it passes. For a given temperature and density of air vapor mixture the rate of vapor travel by molecular diffusion between two points may be considered as directly proportional to the difference in vapor pressure and inversely proportional to the distance between the points. Thus under ordinary conditions it may be substantially correct to assume that the laws governing the rate of vapor travel through air by either convection currents or molecular diffusion are similar in form to those governing the flow of heat through air by conduction and convection.

When vapor passes through a solid material there are at least three types of materials which seem to be of importance. First, those materials which are permeable to air or gas and which will not absorb water vapors; second, those materials which are impermeable to gas but which will absorb water vapor; and third, those materials which are permeable to gas and which will also absorb water vapor. Temperature is also an important factor.

If a non-hygroscopic material is permeable to water vapor and its temperature at all parts is above the dew-point temperature of the vapor in contact with those parts, the vapor will travel through the material without change in state. If the static pressure on the two sides of the material is balanced, the vapor transmittance should be by molecular action and the rate of transfer should be proportional to the vapor pressure difference and inversely proportional to the distance of travel. The

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unit vapor resistance of the material should depend partly upon arrangement of cells or open spaces through which the vapor may travel, and it should be higher than the unit resistance for still air. If the temperature at some point along the path is below the dew-point temperature of the vapor at that point, condensation may occur, forming either water or ice depending upon the temperature. The uniform transfer of vapor will be disturbed, new vapor pressure gradients will be established, and there will be a tendency to accumulate moisture within the material.

If a material which is impermeable to water vapor is hygroscopic, it will condense water vapor and establish a moisture equilibrium when in contact with air vapor mixtures. The percentage of moisture absorbed by the material will depend somewhat on the temperature, but largely upon the relative humidity of the air with which it is in contact. The transmission of moisture through materials of this character will be by capillarity or some similar process and will not depend upon vapor pressure difference. Under certain conditions it appears that vapor may be absorbed from a mixture of low vapor pressure, transmitted through the material, and delivered to a mixture of high vapor pressure. If the temperature of either surface of the material is below the dew-point temperature of the vapor in contact with it, some of the vapor will be condensed forming either free water or frost, depending upon the temperature. Free moisture or water may disturb the normal moisture equilibrium which would have been established by the water vapor in contact with the material and thus change the rate, and perhaps direction, of moisture flow through the material.

If a material is hygroscopic and permeable to water vapor, it may transmit water vapor and free moisture, the net result depending upon several factors. For instance, a condition might be set up in which vapor would be transmitted in one direction due to vapor pressure differences and the permeability of the material, and moisture would be transmitted in the opposite direction due to the hygroscopic properties of the material and the relative percentages of vapor saturation on the two sides of the material.

The wall of Fig. 1 is built of a homogeneous, non-hygroscopic material which is permeable to water vapor. The air in contact with the left hand surface of the wall is at 70 F and 40 per cent relative humidity, and that in contact with the right hand side is at 0 F and 40 per cent relative humidity. The temperature gradient line through the wall is plotted to the temperature scale shown at the left, and the vapor pressure lines are drawn to the vapor pressure scale shown at the right. In each instance a reasonable drop in temperature and vapor pressure is indicated between the surface of the material and vapor in contact with this surface.

Under the conditions given the line *A-B*, plotted to the temperature scale, represents the temperature gradient through the wall. The curved line *C-D*, plotted to the vapor pressure scale, represents the maximum vapor pressure that it would be possible to carry at any point within the wall without condensation. In other words any point on this curve gives vapor pressure at saturation for the temperature of the material in the wall at that point. The straight line, *E-F*, is the normal vapor pressure gradient established in the wall due to the vapor pressures on each side of the wall. In this case the line *E-F* does not cross the curved line *C-D*, indicating that the temperatures at all points in the wall are above the dew-point temperature of vapor at corresponding points. The vapor should thus pass from *E* to *F* through the wall without condensing within the wall.

The wall shown in Fig. 2 is identical in every respect with that shown in Fig. 1, and the conditions as to surrounding temperatures and vapor pressures are the same with the exception that the relative humidity of the air

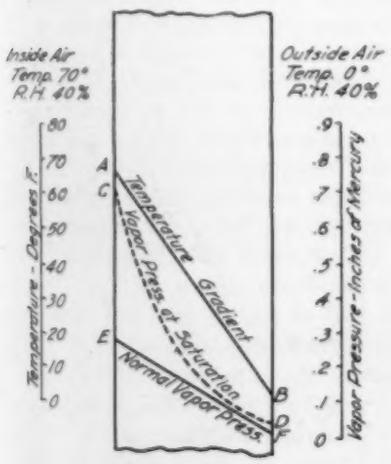

Fig. 1—Wall built of non-hygroscopic, homogeneous material permeable to water vapor. Curved line *C-D* represents maximum vapor pressure that can exist in wall without condensation. Line *E-F* represents normal vapor pressure established by air conditions on two sides of wall. Condensation will not occur

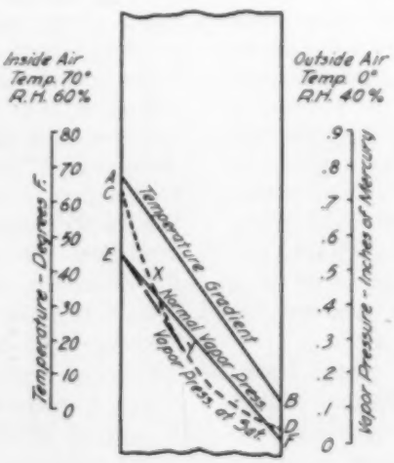

Fig. 2—Wall built of non-hygroscopic, homogeneous material which is permeable to water vapor. Line *E-F* represents normal vapor pressure established by atmospheric conditions on two sides of wall. Since *E-F* crosses maximum vapor pressure line *C-D* condensation will occur. *E-Y-F* represents actual vapor pressure line established in wall

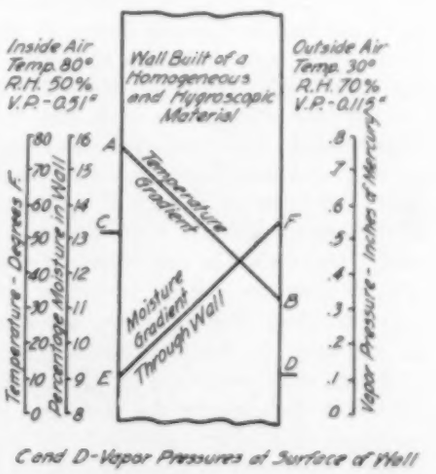

Fig. 3—Wall built of homogeneous, hygroscopic material impermeable to water vapor. Moisture will travel through wall in reverse direction to vapor pressure drop between two surfaces of wall

on the left hand surface of the wall has been raised from 40 to 60 per cent, thus increasing the pressure of the vapor in contact with this surface. In this case the vapor pressure line, *E-F*, which would be established through the wall due to vapor pressure conditions in contact with the two surfaces will be much steeper than that for Fig. 1, and as shown will cross the limiting vapor pressure line *C-D* at some point *X* within the wall. This means that the temperature of the material at *X* will be below the normal dew-point temperature of the vapor at this point and condensation should take place. Since the temperature of the material in the wall at points immediately beyond *X* is such that the maximum vapor pressures are below those which would normally be established, the rate of vapor travel will be accelerated and new vapor pressure gradients will be established through the wall. If the straight line, *E-Y*, is drawn tangent to the curve, *C-D*, at *Y*, it seems probable that *E-Y* would represent the vapor pressure gradient established in the wall up to the point *Y*. From this point on the maximum vapor pressure gradient for each successive point would be established by a tangent to the curved line, *C-D*, and when the slope of this tangent is less than the slope of the line *E-F*, the rate of vapor travel would be less than the normal rate. The rate at which moisture enters the wall would thus be increased and the rate at which it leaves would be decreased from the normal, and there would be an accumulation of moisture within the wall from the point *Y* to the outer surface. Since in this case the interior temperatures of the wall would be below 32 F, frost or ice would be formed.

The wall of Fig. 3 is built of a homogeneous hygroscopic material which is impermeable to water vapor. The air on the left hand side is at 80 F and 50 per cent relative humidity, and that on the right hand side is at 30 F and 70 per cent relative humidity. The normal temperature gradient through the wall is shown by the straight line *A-B* plotted to the temperature scale at the left. The pressure of the water vapor in contact with the two surfaces is shown by the two short heavy lines, *C-D*, and plotted to the vapor pressure scale on the right. This represents the vapor pressure difference on the two sides of the wall, but, since the material is impermeable to vapor, the vapor cannot pass through as such and therefore there is no connecting vapor pressure line between *C* and *D*. Materials of this nature absorb water directly from the vapor with which they are in contact. The percentage of water which they will absorb does not depend upon the absolute vapor pressure but does depend on the per cent of saturation or, in this case, the relative humidity of the air.

From data published by the United States Forest Products Laboratory the moisture equilibrium content of wood in contact with 80 F, 50 per cent relative humidity air would be approximately 9 per cent by weight and, when in contact with air at 30 F and 70 per cent relative humidity, it would be slightly over 13 per cent by weight. Thus the line *E-F* shows the probable moisture gradient through the wall which would be established by the air conditions shown. From this it is evident that the moisture travel through this wall would be from right to left, or from the side in contact with the low vapor pressure air to that in contact with high vapor pressure air. This is contrary to the usual assumption that moisture travels through a wall in direct proportion to the vapor pressure drop between the two sides of the wall. It should be remembered, however, that in this case the vapor does not pass through the wall but that it is condensed and absorbed by one surface of the wall, transferred through as free moisture, and evaporated from the other surface. The direction of vapor travel is determined by the moisture equilibrium conditions for the two sides of the material and not by the absolute vapor pressure differences. Wood is a very common example of a hygroscopic material used in buildings which has a very low permeability to vapor. In most cases the sheets of wood are not continuous and vapor may travel through the openings by diffusion or convection.

In most practical cases where the vapor pressures are different on the two sides of a building wall a vapor pressure gradient will be established through the wall, and the vapor will travel in the direction of vapor pressure drop. The slope of the vapor pressure line for different sections of a wall will depend upon the resistance to vapor passage at various parts of the wall. Figs. 4 and 5 represent two walls which are built of homogeneous non-hygroscopic materials which are permeable to water vapor. Both are subjected to air on the warm side at 70 F and 60 per cent relative humidity, and air on the cold side at 0 F and 40 per cent relative humidity. The inside or warm surface of Wall 4 is lined with a material which has a high resistance to vapor penetration, and for Wall 5 this vapor resisting material has been applied to the outer or cold surface of the wall. The normal vapor pressure gradient which would be established by the vapor pressures on the two sides of the walls is shown by the broken lines *E-F*. Since the wall of Fig. 4 has a high vapor resistance on its warm surface, the vapor pressure within the wall is comparatively low, but in Wall 5 with the vapor resistance on the cold surface the vapor pressure within the wall is high. If the temperatures throughout both walls were at all points above the dew-point temperatures of the vapor at these points in the wall there would be no condensation. For the temperature conditions shown, however, the high vapor pressures within Wall 5 are above the maximum allowable vapor pressures throughout the greater part of the wall and moisture will be formed. In Wall 4 the vapor pressures are below the danger point throughout the wall and no vapor will condense. From the discussion of the condition shown in Fig. 2 it is evident that the line *E-F* of Fig. 5 does not represent the true vapor pressure gradient established in the wall. The actual line cannot be above the curved line *C-D*.

Fig. 6 represents a wall built of two materials, *A* and *B*, both of which are permeable to water vapor. If the vapor resistance of *B* is twice that of *A*, and if the horizontal lines *C* and *E* represent to scale the vapor pressure on the corresponding sides of the wall, then the broken line, *C-D-E*, represents the normal vapor pressure gradient that would be established through the wall due to the vapor pressures on the two sides. If the temperature at the central section of the wall, *X-Y*, is high enough to allow the vapor pressure, *D*, without con-

densation, then the vapor will be transmitted at the same rate through all parts of the wall and there will be no condensation within the wall. If the temperature at the central section of the wall is lowered until the maximum vapor pressure that can be carried is at some lower point, *F*, then the vapor pressure line, *C-F-E*, will be established and evidently the rate of vapor travel through material *A* will be increased and that through material *B* will be decreased and there will be an accumulation of moisture within the wall.

If the vapor pressure on the left side of the wall is decreased to a point C_1, which is 50 per cent of that at *C*, and which will result from lowering the relative humidity of the air in contact with this surface to 30 per cent without changing its temperature, then the pressure gradient through material *A* would be *C'F* and the amount of moisture transferred to the central plane of the wall would be decreased, but since the vapor pressure at *F* is still determined by the same wall temperature as previously the vapor pressure gradient *F-E* is not changed and therefore the rate of vapor travel from the central section to the outer part of the wall is not changed. It is evident that there will be a reduction in the rate of accumulation within the wall and the percentage of this reduction will be greater than the percentage by which the vapor pressure on the left side of the wall was reduced. Thus, when condensation occurs within a wall, a reduction in the relative humidity on the warm side of the wall will reduce the rate of condensation by a greater percentage than the percentage reduction in relative humidity providing that there is no change in temperature.

Figs. 7 and 8 represent two typical frame walls which are identical with the exception that Wall 8 has been insulated with a fill insulation between the studs. In both cases the air on the warm side is at 70 F and 40 per cent relative humidity, and that on the cold side is —20 F and 40 per cent relative humidity. The temperature gradients through the walls are shown to scale by the lines *A-B;* the maximum vapor pressures that may be carried at any point within the wall for the corresponding temperatures are shown by the dashed lines *C-D;* and the probable vapor pressure gradients established by the vapor pressure differences on the two sides of the wall are shown by the lines *E-F*. The lines *E-F* are shown as curved lines through the central section of the walls as it is probable that the vapor is carried through this section partially by convection and not entirely by molecular diffusion.

The addition of insulation to the wall of Fig. 8 reduced the temperature of the inside surface of the sheathing from approximately 26 F to —6 F. This reduced the maximum vapor pressure which could be carried at this point without condensation. In the wall of Fig. 7 the vapor pressure line *E-F* does not cross the limiting pressure line *C-D*. In the wall of Fig. 8, however, the surface temperature of the sheathing has been reduced to such an extent that the maximum vapor pressure that can be carried on the inside surface of sheathing shown by *G* is less than the vapor pressure *H* required for an equal rate of vapor travel through all sections of the wall. This means that the rate of vapor travel from the inside or warm surface of the wall to the sheathing will be increased as the vapor head, *E-G*, is greater than the vapor pressure head, *E-H*, and that the rate of vapor travel from the surface of sheathing to the outside or cold surface of the wall will be decreased as the vapor head, *G-F*, is less than the normal required vapor head *H-F*.

When vapor accumulates at any point within a wall it is due usually to the fact that the vapor travels through the warm section of the wall to the accumulation area at a greater rate than it travels from this area to the cold side of the wall. If it is assumed that the temperatures are fixed, then the rate of condensation may be reduced first by reducing the rate at which vapor travels from the warm side of the wall to the area of condensation, or second by increasing the rate at which it may travel from this area to the cold side of the wall. The rate at which vapor travels from the warm area to

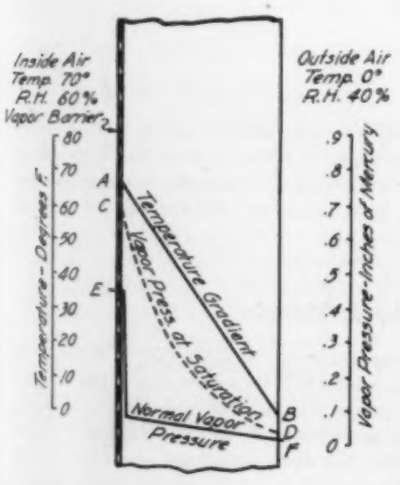

Fig. 4—Wall built of non-hygroscopic, homogeneous material which is permeable to water vapor. Vapor barrier placed on high vapor pressure side of wall. No condensation will occur

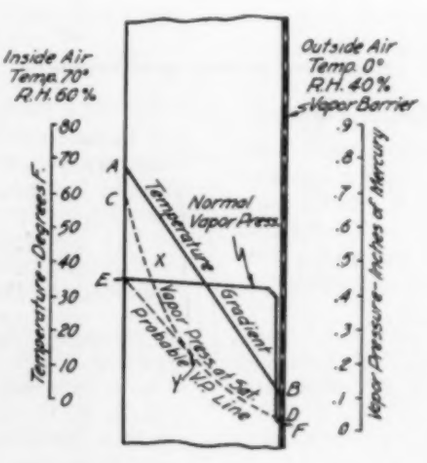

Fig. 5—Wall built of non-hygroscopic homogeneous material permeable to water vapor. Vapor barrier placed on low vapor pressure side of wall. Condensation will occur between points *Y* and *F* through wall

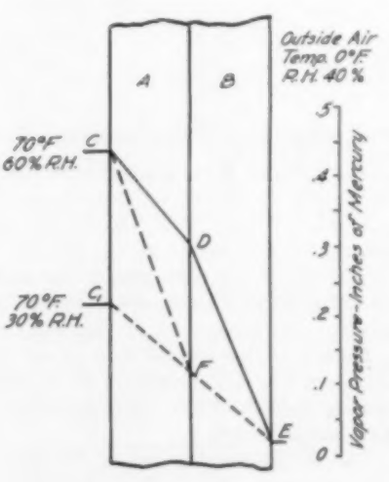

Fig. 6—Wall built of two materials each non-hygroscopic and permeable to water vapor. Vapor resistance of *B* equals two times the vapor resistance of material *A*

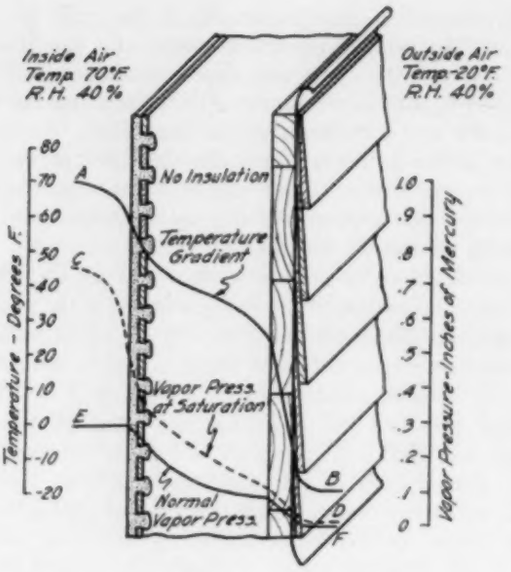

Fig. 7—Frame wall without insulation showing normal temperature and vapor pressure gradient through wall

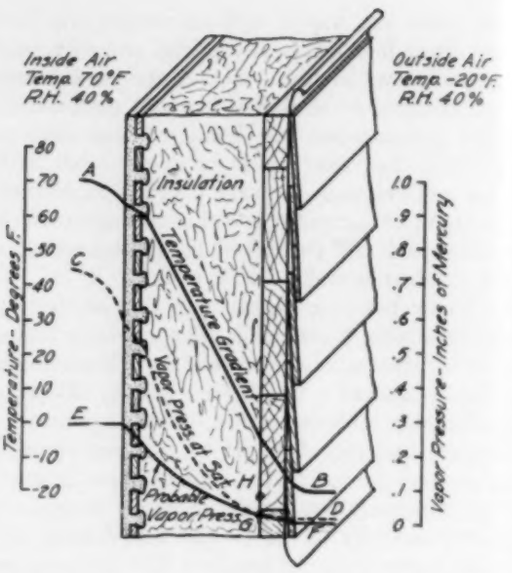

Fig. 8—Frame wall with insulation showing temperature and vapor pressure gradients as established through wall. *H* indicates vapor pressure required at inner surface of sheathing to prevent condensation. *G* indicates maximum vapor pressure allowed by temperature of sheathing

the inner section of a wall may be reduced first by reducing the relative humidity and thus the vapor pressure on the warm side of the wall, and second by introducing some material with high vapor resistance on the warm surface of the wall. The rate at which vapor may travel through the outer section of a wall may be increased by reducing the relative humidity on the outside of the wall, (this is usually impractical or impossible), or by constructing a wall with outer surface materials which have low resistance to the passage of vapor. One of the most effective methods for reducing the moisture accumulation within a wall of a building is to reduce the relative humidity of the air in contact with the warm surface of the wall.

In the past it has been common practice to use a building paper or some similar material on the outside surface of a wall to protect it from free moisture in contact with the outside surface and to reduce air leakage through the wall. Recently many of the building papers have been improved and are now good vapor barriers, thus preventing the escape of moisture which may pass through the inside surface into the wall.

All of the laws governing the flow of vapor through materials have not been demonstrated by experiments, but data available indicate that in many cases the flow of vapor through a material or combination of materials is proportional to the vapor pressure drop along the path of flow.

From the nature of certain materials and from the fact that vapor may be condensed either by lowering the temperature or by the hygroscopic property of many materials, it is evident that there will be many different conditions under which the flow of vapor may not follow any simple law.

Propeller Fan Manufacturers Association Formed

The *National Association of Fan Manufacturers* have recently formed the *Propeller Fan Manufacturers Association* with headquarters at Detroit, Mich. It has been announced that the reason for this sponsorship is due to the importance now being given to the use of propeller fans in the field of ventilation and the need for an association where manufacturers with identical interests could function along that channel.

Officers elected for the term of one year are: M. W. Bauer, Aerovent Fan Co., president; A. R. Stephan, DeBothezat Ventilating Equipment Div., vice-president; and V. C. Shetler, secretary and treasurer.

Charter members consist of Aerovent Fan Co., Air Controls, Inc., American Blower Corp., American Coolair Corp., Autovent Fan & Blower Co., Buffalo Forge Co., Chelsea Fan & Blower Co., DeBothezat Ventilating Equipment Div., Hartzell Industries, Inc., Ilg Electric Ventilating Co., International Engineering, Inc., Marathon Electric Mfg. Corp., Peerless Electric Co., Propellair, Inc., and Viking Air Conditioning Corp.

A certified rating label has been adopted which will be issued only to those member companies whose ratings and tests are in accordance with the Standard Test Code for Centrifugal and Propeller Fans adopted by the *National Association of Fan Manufacturers.*

New Firm of Herske & Timmis

Announcement has been made of the formation of the firm of Herske & Timmis, Inc., which will introduce a new line of controls, systems and equipment for heating and air conditioning, with A. R. Herske as president and W. W. Timmis as vice-president. Miles Gordon has been appointed manager of the concealed radiator division and Edward Walton is chief engineer. The main office is located at 33 West 60th St., New York, N. Y.

Distribution in major cities east of the Mississippi will be carried out through local agency representatives. All of the products have been designed to be correlated with one another and emphasis will be given to system sales.

Evaluating Performance of Water Heaters Fired with Solid Fuels

By H. J. Rose,* (*MEMBER*) and R. C. Johnson,** Pittsburgh, Pa.

MORE than 200,000 coal and wood water heaters for domestic hot-water supply are produced annually in this country. This is more than the number of house-heating boilers that are produced,[1] so that methods for evaluating the performance of such equipment should be of interest to all who design, select, install and operate such equipment.

Water heaters are commonly rated and sold in terms of the output of hot water produced at their full burning rate. The householder is also interested in such factors as the amount of attention required, uniformity of water temperature, length of time that the fire will bank without dying out, and economy. However, there is a lack of published methods for testing solid-fuel-fired water heaters so that they can be compared in terms of consumer satisfaction.

Several years ago the writers† were engaged in the engineering development of water heaters having improved performance characteristics. In order to evaluate the development models, and to compare them with existing types, it was necessary to devise test methods. These measured not only the output at continuous full burning rate, but also flexibility of output and response to automatic controls, banking characteristics, amount of attention required, performance of heater and storage tank in meeting simulated household demands, and other factors.

After the heaters were tested by these methods in the engineering laboratory, they were placed in homes, and instrument records were kept of their performance over a considerable period of time. In this way it was found that the tests described in this paper give a reliable index of the performance that can be expected in actual household use.

Rate of Hot-Water Output

All water heaters, regardless of the kind of fuel used, may be classified into three general types with regard to flexibility of hot-water output as follows:

a. Instantaneous heaters—Require no storage tank as the water is heated as fast as it is drawn.

b. Quick-recovery heaters—Used in connection with storage tanks, the size of which depends upon the flexibility of hot-water production and the maximum quantity of water required during periods of demand.

c. Slow-recovery heaters—Have a low maximum hourly output and require either a large storage tank in which to accumulate a liberal reserve supply of hot water, or else a low demand rate.

There is no sharp dividing line between quick-recovery and slow-recovery water heaters, and the methods given can be used for testing both types.

This paper does not attempt to set up performance specifications or standards since this is a function of appropriate standardizing groups. Instead, it presents a correlated series of tests which give rather complete information on the performance of water heaters, both by themselves and in connection with storage tanks. The information obtained is particularly valuable for giving designers and manufacturers an accurate idea of heater performance. For some purposes, only a part of the tests need be made.

Apparatus Required

Fig. 1 is a schematic arrangement of the set-up used. All of the tests can be made with this equipment, although not all of it is needed for each test. The heater is shown on platform scales with flexible water connections and a sliding stack connection. This permits the fuel consumption to be determined at frequent intervals. An alternate method eliminates the scales and flexible connections, in which case the fuel consumption during a

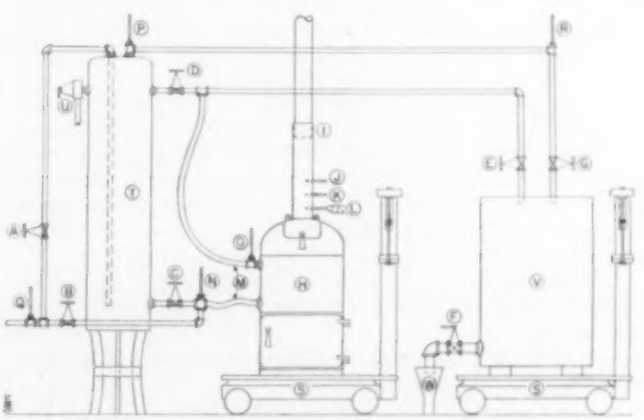

Fig. 1—Set-up for testing solid-fuel-fired water heaters

Key to Fig. 1

A, B. Valves on cold-water inlets to storage tank and heater.
C, D. Gate valves of ¾ in. or larger size, depending on heater output.
E. Needle valve with graduated wheel for close adjustment of flow during output tests.
F. Quick-opening gate valve 1½ in. or larger, for emptying weighing tank.
G. Gate valve used for draw-off tests.
H. Hot-water heater.
I. Stack with slip joint to permit weighing heater.
J. Gas-sampling tube.
K. Draft tube.
L. Thermocouple or thermometer for determining stack temperature.
M. High-pressure flexible connections to permit heater to be weighed.
N, O, P. Thermometers, preferably of recording type.
Q, R. Thermometers.
S. Scales for weighing heater and weighing tank.
T. Hot-water storage tank of selected size.
U. Temperature and pressure relief valve piped to drain.
V. Weighing tank.
W. Drain.

Note: Draft dampers and thermostatic control are not shown in the drawing, since these vary widely in type and location.

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**Industrial Fellow, Anthracite Fellowship, Mellon Institute of Industrial Research, and† formerly Research Chemist, the Philadelphia and Reading Coal and Iron Co., Pottsville, Pa.

[1] Census of Manufactures: 1937, Heating and Cooking Apparatus, Except Electric. (*U. S. Bureau of the Census, Industry Release No. 1119*, Dec. 6, 1938.)

For presentation at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Mackinac Island, Mich., July, 1939.

test is determined by shaking the grates and filling to the original level with a weighed amount of the fuel.

Stack Drafts

A definite stack draft is maintained in all of the methods described, although in actual household use, heaters are subject to variable natural draft conditions, which are influenced as much by the chimney and by weather conditions as by the characteristics of the heater. Since natural chimney draft fluctuates too much to be very suitable for comparative engineering tests, it has been found preferable to apply a definite draft to the heater.

It was found that a draft of 0.03 in. water is suitable for practical testing, but tests have been made at drafts of 0.01, 0.05 and sometimes 0.07 in. of water.

Maximum Output

This test measures the hourly rate of hot-water production from a running start, when the fire is burned at a definite applied draft. It does not differ materially from some methods that are in use for determining the hot-water output for rating purposes. Only the general procedure will be outlined, since it is not the purpose of this paper to discuss the details of testing technique.

1. By-pass the hot-water storage tank and provide a slow continuous flow of water through the heater and the weighing tank to the drain. This is done by closing valve *A*, opening *B*, closing *C* and *D*, opening *E* slightly, and opening *F*.

2. Start a fire in the heater with paper and charcoal or other kindling, and fill with the solid fuel which is to be used in the test.

3. Apply a fixed draft of 0.03 in. water, measured in the stack close to the breeching.

4. Adjust valve *E* as often as necessary to maintain a temperature rise of 100 F in the water passing through the heater. When the heater appears to have reached its full output, shake the grates until red coals appear, remove ashes, fill the heater with fuel, record the weights of heater and weighing tank *V*, close valve *F* and record the time of starting the test.

5. Continue to maintain a temperature rise of 100 F through the heater by adjusting valve *E*, and weigh the water discharged at 15- or 30-min. intervals. When the weighing tank is full, empty it by means of the large quick-opening valve *F*, noting the time required for drainage, so that a correction can be made for the inflow during the time that valve *F* is open. (An alternate method would be to provide two weighing tanks.)

6. If the heater is on scales *S*, as shown in Fig. 1, the weight should be recorded at hourly intervals to determine rate of coal consumption.

7. Obtain stack temperatures throughout the test, at *L*, preferably with a recording instrument.

8. It is desirable to obtain stack gas analyses.

9. Continue the test until the heater needs refueling as determined by the condition of the fire. If it is on scales, record the loss in weight. Shake the grates until red coals appear, remove the ash, and fill the heater with fresh fuel to the original level, and reweigh. (If the heater is not on scales, the amount of fuel burned is determined from the weight of fuel required to refill.) The carbon content of the ash should be determined. The heater is now ready for the pick-up test procedure given in par. 10.

The output per hour in gallons of water of 100 F rise should be plotted against time. A secondary scale showing the total Btu introduced into the water per hour may be provided on the graph. Efficiency should be reported as the percentage of the fuel's heat content which appears in the hot water. The gallons of 100 degree-rise water produced per pound of fuel may also be reported.

The test shows the output of the heater hour by hour, over the entire period from fueling to refueling. Commercial ratings of water heaters vary with respect to the temperature rise used and the length of time over which the output is averaged. For engineering comparisons, complete data are desirable.

Pick-up of Fire Which Has Required Refueling

The maximum output test described was made with a running start, whereas this test measures the rate of pick-up of a fire which has needed refueling and has had the maximum amount of fresh fuel added. This simulates a common household condition.

10. Start with the freshly tended fire described in par. 9, and continue the use of 0.03 in. water draft and 100 deg rise, weighing the water at 15-min intervals until full output is reached.

Rate of Decrease of Output When Fire Is Checked

As soon as the heater reaches full output in the pick-up test, the fire is checked, and the rate of decrease in output is determined. This measures any tendency for an overrun in storage-water temperatures, and the manner in which the heater will respond to control.

11. Starting with the heater producing its full output, as obtained in par. 10, damper off the fire to the fully banked position without changing the stack draft, and determine the output by the same procedure at 15-min intervals until a fairly constant low output is reached.

Minimum Output

As soon as the low banking rate has been reached, the heater is left in this banked condition over night. If desired, the output obtained during this period may be determined, but this is not always done since the actual outputs at the beginning and end of banking are regularly obtained.

12. As soon as a low banking rate has been obtained according to par. 11, the heater is left in a banked condition for 12 hours without attention and without changing the rate of water flow. If the output is not determined during this period, it should be determined for one hour at the end of the test.

Pick-up After Banking

This test measures the responsiveness of the heater when the dampers are opened after a long banking period without giving any attention to the fire.

13. Continuing from par. 12, set the dampers in the full burning position but do not shake the grates or add coal. Continue the use of 0.03 in. water draft, maintain a 100 deg temperature rise, and determine the output at 15-min intervals, until a satisfactory output is obtained, or until more fuel is needed (as determined by the condition of the fire).

Graph the results of the tests covered by pars. 10 to 13 inclusive on a single sheet, plotting gallons of 100 degree-rise water against time. A supplementary Btu scale may be included on the graph. The results of the foregoing tests will give an accurate idea of the maximum and minimum outputs of the heater, and its response to damper control at the stated draft with the particular fuel used.

Length of Time That Heater Can Be Banked

It is desirable for water heaters to be able to remain banked for several days without danger of the fire dying out. This permits the family to be absent over the week-

end, for example, and to have a tankful of hot water and the fire in good condition upon returning. The water should not overheat during banking periods, for several important reasons.

Inexpensive thermostatic damper controls are available for solid-fuel-fired water heaters. If the heater does not come equipped with such a control, one should be installed for the remaining tests.

In this test, the heater is connected with the minimum recommended size of storage tank, and the fire banked under thermostatic control, without drawing off any water, and without giving any attention to the heater until more fuel is needed. The purpose of this test is two-fold. It measures the maximum length of time which the fire can be banked without attention, and shows whether there is any tendency for the water to overheat in a minimum size tank, when no water is drawn off.

The authors have tested small water heaters burning anthracite fuel, which could be banked continuously for four days or more without danger of the fire dying out, and without any tendency to overheat the water in a 30 gal, fully insulated tank. Larger heaters of similar design have been banked for 10 days or more with the same results. At the end of these long banking periods, the fire was still in good condition. This indicates what can be expected from solid-fuel-fired water heaters which have been carefully designed on the basis of reliable test data.

14. As soon as the test described in par. 13 is completed, shake the grates, remove the ash, fill the heater with fuel, weigh, and connect it to the storage tank which is filled with cold water. This storage tank should be of the minimum size which is recommended for use with the particular heater, and should be provided with the maximum insulation (if any) which is recommended. The change-over to the tank is made by opening valve *A*, closing *B*, opening *C* and *D*, and closing *E*.

An appropriate thermostatic damper control should be used. This control keeps the damper in the burning position until the tank is filled with hot water, after which it will function from time to time to keep the water temperature up to the desired level. Temperatures at the top of the hot-water storage tank should be recorded throughout, preferably by a recording thermometer.

No hot water should be drawn off and no attention whatever given to the fire until the end of the test, which occurs when more fuel is needed, as determined by the condition of the fire.

15. Determine the weight of fuel consumed. Shake the grates until red coals appear, remove ash, and fill with fuel.

Report the length of time that the heater can be banked under these conditions; the amount of fuel consumed; and graph the temperature at the top of the hot-water-storage tank throughout the test.

Tests Simulating Household Conditions

All water heaters (except those of the instantaneous type) are used with storage tanks. Since satisfactory performance under typical household conditions depends to an important degree upon the size and location of the tank, the damper controls, etc., as well as upon the water heater itself, it is essential to determine the actual performance of a particular combination that is recommended for general use.

Hot-water demands in private homes vary greatly from hour to hour, being characterized by heavy draw-offs at certain times, and periods of many hours when there is little or no demand. This increases the difficulty of designing hot-water heaters as compared with the design of hand-fired house-heating boilers where the changes in load are more gradual.

Table 1—Draw-Off Schedule for 50 Gal Demand[a]

Hour of Draw-Off	Water Gal of 100 F Rise	Btu in Draw-Off
8 a.m.	15	12,255
9 a.m.	6	4,902
10 a.m.	3	2,451
11 a.m.	1	817
12 m.	3	2,451
1 p.m.	9	7,353
2 p.m.	3	2,451
3 p.m.	2	1,634
5 p.m.	8	6,536
	50	40,850

Repeat the schedule on the following day.

[a] *Note:* This draw-off schedule has been condensed to nine hours, whereas hot-water demands in the typical home spread over about 16 hours. This condensed schedule has been used because it fits in well with the regular testing day, and it has been found practical in comparing and evaluating hot-water systems. This schedule is of course more severe than drawing off the same quantity of water over a period of 16 hours, and it thus represents more nearly the rate of demand on peak days.

For simulated household draw-off and banking tests, the heater is connected with a tank of recommended size, and a thermostatic damper control is used. Widely varying amounts of hot water are drawn off hour by hour, according to definite schedules based on hot-water demands in typical homes. The draw-off schedule is continued without any attention being given to the fire until the hot-water system fails to meet the scheduled demand.

Average hot-water demands per family range from less than 50 gal to more than 200 gal per day, depending upon the size of family, ages and personal habits of the members, number of hot-water outlets, number of servants, standard of living, etc. However, even in the case of families with low average consumption, there are periods of peak demand. Storage tanks of less than 30-gal capacity do not provide enough hot water at one time for certain household uses, regardless of the type of heater or the fuel used (whether solid, liquid or gaseous).

Nevertheless, such a large proportion of the tanks sold are of the 30 gal size that heaters have been regularly tested connected to tanks of this capacity as well as larger sizes. Any heater which gives acceptable results when connected with a 30 gal tank will give much better performance with a larger tank, with regard to availability of hot water, avoidance of overruns in water temperature during banking, etc.

The combined draw-off and banking tests are made as follows:

16. The test is started from conditions existing at the end of par. 15, after the fire has been tended and as soon as the water in the tank is at the desired storage temperature. The thermostatic damper control is left in operation, and a stack draft of 0.03 in. water maintained.

17. Hot water is drawn off through valve *G* in accordance with the 50 gal schedule given in Table 1. Since the water will not always have a 100 deg rise in temperature, the actual quantity to be drawn off must be calculated. The most convenient method is to draw off a weight of water at the observed temperature which will contain the desired number of Btu. In Table 1 it is assumed that within the temperature range used, each gallon of water contains 8.17 Btu per degree temperature rise. The most convenient method is to draw off a weight of water at the observed temperature which contains the specified number of Btu. The temperature of the water discharged is observed continuously during the draw-off, by thermometer *R*. If the temper-

ature changes much, the water is weighed in increments so that the heat content can be accurately calculated. The temperature of incoming cold water Q must also be recorded during this test.

18. The 50 gal draw-off schedule given illustrates how it applied to a particular laboratory working day.

19. Larger draw-offs may be made, especially with heaters or tanks of higher capacities. In this case proportional amounts are withdrawn; for example, a 100 gal schedule is obtained by doubling the quantity of water drawn off each hour in the 50 gal schedule.

20. At the end of each day's scheduled draw-off, the heater is left under thermostatic control until the following morning, when the same schedule is repeated. This cycle is continued until the system is unable to meet the full scheduled demand with hot water of usable temperature (*e.g.*, 100 F or more). No attention is given to the fire at any time after starting the test. Determine the amount of fuel consumed.

In reporting these tests, the following should be given: The number of days during which the 50 gal or 100 gal (or larger) schedule was met without any attention to the heater, the total gallons of hot water of specified temperature rise which were drawn off during the test, and the pounds of fuel consumed. It is convenient to report efficiency as gallons of 100 degree-rise water produced per pound of fuel burned, during the entire test. The combined draw-off and banking test, as described before, forms an important practical method for evaluating the over-all performance of a water heater and storage tank. It is often desirable to make this test first, so that a good idea of characteristic performance and limitations can be obtained, before the more specific tests are authorized.

Summary

This paper briefly describes methods by which solid-fuel-fired water heaters (for domestic hot-water supply) may be tested for maximum and minimum output, flexibility of output or response to control, maximum attention intervals and efficiency. A combined draw-off and banking test is described which simulates household demands, and gives a good idea of the over-all performance which can be expected in actual use.

Field installations of water heaters based on knowledge gained by these tests have proved so satisfactory, in comparison with previous methods of water heating used in the same homes, that these test methods should be of engineering interest.

Oregon Members to Form Chapter

May 8, 1939. The first meeting of the Oregon group of ASHVE members was called to order by J. D. Kroeker, chairman pro tem, in the presence of C. E. Heinkel, acting secretary, and 22 members and candidates for membership in the ASHVE.

After a brief introductory talk on the subject of organization, Chairman Kroeker called on B. W. Farnes for a resumé of his visit to the Pacific Northwest Chapter at Seattle on the occasion of Pres. J. F. McIntire's address.

Chairman Kroeker then read the suggested petition for a charter for the Oregon group and, in order to make the petition complete, it was necessary to choose a name. C. E. Enders proposed the name, Columbia Empire; E. C. Willey suggested Oregon Chapter; and Mr. Farnes proposed the name, Far Western. Following a discussion, a vote was taken and resulted in the selection of Oregon Chapter. It was then moved by B. W. Moore and seconded by Walter Hanthorn that the petition be accepted as read with the name, Oregon Chapter, inserted, and this motion was carried.

Announcement was made by Chairman Kroeker that arrangements had been made for William Goodman, Trane Co., to address the group on August 30. Upon motion, it was voted that a committee be appointed to choose a topic for Mr. Goodman's talk. T. E. Taylor was named chairman of this committee and J. P. McDermott and E. S. Turner agreed to serve as members.

The preamble of the proposed By-Laws was read by Mr. Kroeker and during the discussion which followed, it was decided to elect a committee of seven to consider the By-Laws, revising them to meet the requirements of the proposed Oregon Chapter. A ballot was taken and resulted in the election of the following committee: T. E. Taylor, chairman; H. K. Mead, E. C. Willey, C. E. Heinkel, W. J. Kollas, B. W. Farnes and J. D. Kroeker.

Mr. Taylor suggested that since he had been chosen chairman of two committees, it might be well to appoint a new chairman of one and relieve him of this responsibility. This was agreed upon and Cecil MacGregor was chosen as the new chairman of the committee to select a topic for Mr. Goodman's talk.

Professor Willey reported on the results of the poll taken of people attending the Second Air Conditioning Conference at Oregon State College. He then outlined some of the research work being done at the college and told of plans for building an experimental house to further their research studies.

Since there was no additional business, the meeting was declared adjourned by Chairman Kroeker.

Delco-Heat Returns to Rochester

Arrangements for the sale of all Delco-Heat Products under the management of the Delco Appliance Division, General Motors Sales Corporation, Rochester, N. Y., after having been handled through the Delco Frigidaire Conditioning Division, Dayton, Ohio, since January 1, 1936, are now in effect according to E. A. Halbleib, general manager of the Delco Appliance Division.

C. E. Lewis, formerly sales manager for Delco-Frigidaire, heads the new sales organization. He has been well-known in the automatic heating and air conditioning business for many years and his broad experience in engineering and production, as well as in sales and distribution, make him ideally fitted for his new work.

Advertising and sales promotion is under the direction of O. E. Wolf, who occupied the same position with Delco-Frigidaire. George Johnson, for many years service manager of Delco Appliance Division, will be service manager on Delco-Heat. All engineering and product development will continue under the direction of W. H. Hutchins.

Pres. McIntire Attends Heating Association Meetings

Pres. J. F. McIntire officially represented the Society at the Fiftieth Anniversary Meeting of the *Heating, Piping and Air Conditioning Contractors National Association*, Chicago, Ill., May 31-June 2, at which the following officers were elected: President—W. J. Olvany, New York, N. Y.; vice-president—J. E. McNevin, Denver, Colo.; treasurer—S. A. Pope, Chicago, Ill.; board of directors—G. P. Nachman, Cleveland, O.; Daniel Hayes, San Francisco, Cal.; L. F. Hudepohl, Cincinnati, O.; W. H. Driscoll, New York, N. Y., A. F. Nass, Pittsburgh, Pa. Members of the board of directors who were reelected are: Thomas O'Callaghan, Boston, Mass., S. A. Pope, Chicago, Ill., and G. J. Stoehr, Milwaukee, Wis. J. C. Fitts, New York, N. Y., was reappointed secretary.

President McIntire was a speaker at the mid-year meeting of the *National Warm Air Heating and Air Conditioning Association*, held at the Stevens Hotel, Chicago, Ill., June 5-7.

Effect of Heat Storage and Variation in Outdoor Temperature and Solar Intensity on Heat Transfer Through Walls

By J. S. Alford,* Lynn, Mass., J. E. Ryan,** Schenectady, N. Y., and F. O. Urban,*** Bloomfield, N. J.

Introduction

AN accurate estimate of either the heating or cooling load is essential to an economic selection of air conditioning equipment. These loads are influenced by five basic items, namely:

1. Heat transmission through walls, windows, etc., resulting from a difference between the temperature of the air on one side of the partition and the temperature of the air on the other side.
2. Radiant energy of the sun absorbed by the outer surface of an exposed wall or transmitted directly through glass partitions.
3. Sensible heat and moisture of outside air introduced to the conditioned space either by infiltration or by controlled ventilation.
4. Sensible heat and moisture of occupants.
5. Sensible heat and moisture liberated by lights, electrical equipment, gas-burning equipment and other miscellaneous sources.

The loss of heat from a structure in winter is occasioned by items one and three only. Hence these are the only ones considered in determining the maximum heating load. The other three items are heat gains, or deductions from the heat loss, and may be of importance in the design of the system and its control.

The heat gain of a structure in summer includes all five items listed. Hence, each one must be considered carefully in calculating the maximum refrigeration requirements for summer air conditioning. In certain structures heat transmission through walls and radiant solar energy transmitted directly through glass partitions account for a large share of the total cooling load. The transmission of heat through walls exposed to the sun is greatly increased because of the radiant energy absorbed by the outer surface of the wall. For example, it is not uncommon for the temperature of the outer surface of a wall exposed to the sun to rise 15 to 30 F above the outdoor air temperature, and even more. Temperatures of flat roofs have been observed as high as 180 F.

A previous paper[1] summarizes the entire heat gain calculation and presents a rational method for determining the increase in heat transmission rate through a wall or roof resulting from exposure to the sun.

Another factor which has been recognized for some time as having a material bearing on the magnitude and time of occurrence of the maximum cooling load is the heat storage capacity of the wall structure. In 1930 Stevenson and Bojer[2] analyzed the transient flow of heat through refrigerator box walls, and obtained a striking agreement between calculated and test results. Two years later, Houghten, Blackshaw, Pugh and McDermott[3] investigated the flow of heat through several roof panels as affected by diurnal variations in outdoor air temperature and solar intensity, obtaining equally good agreement between analytical and experimental work. The importance of heat storage capacity upon the cooling load has been further demonstrated in reports to the Society by several authors, in which considerable discrepancies between calculated and measured cooling loads of actual air conditioned structures have been observed. In some of these cases the nature of the discrepancies gives every indication that they result from the influence of heat storage capacity of exposed walls upon the transmission of heat, and of heat storage capacity of interior partitions and furnishings upon the release of radiant energy transmitted directly through windows.

Heretofore, a consideration of the influence of heat storage capacity upon the cooling load has necessitated working from an assumed temperature of the outer surface of the wall by the method given by Houghten et al[3] or else has necessitated rough approximations.[1] The rough approximation is adequate where other elements of the heat gain, such as ventilation air, occupancy, lights, etc., account for a major portion of the total heat gain, but in other cases is undesirable. On the other hand, the outside surface temperature of a wall depends upon a large number of factors, such as:

1. The latitude.
2. The orientation of the wall.
3. The time of day and time of year.
4. The degree of cloudiness, smokiness and the humidity.
5. The nature and color of the outside surface.
6. Heat transmission and heat storage characteristics of the wall structure.
7. The temperature maintained on the inside of the wall.

Therefore, an extension of the analysis is desirable in order to base it upon the more fundamental considerations of inside and outside surface characteristics, internal properties of the wall, inside and outside air temperatures and intensity of incident radiation.

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[1]Footnote references refer to numbers in bibliography.
Presented at the 45th Annual Meeting of the American Society of Heating and Ventilating Engineers, Pittsburgh, Pa., January, 1939.

The present paper presents a practical method for calculating the heat flow into the interior space as a function of the thermal properties of the wall, the surface coefficients and the variable quantities of outdoor temperature and solar intensity when the wall structure is homogeneous. The solution to this problem was worked out by members of an advanced course in engineering.[4] Although the present paper gives no new experimental data, the accuracy of the method may be demonstrated by using data of Houghten et al[3] where the tests include not only measured rates of heat flow but also records of outdoor air temperature and pyrheliometer measurements of solar intensity.

Assumptions

The basic assumptions employed in this analysis are:

1. The wall is of infinite length and height but of finite thickness. The assumption involves uni-directional heat flow, and is sufficiently accurate for any practical case.
2. The wall is homogeneous, which is sufficiently accurate for un-insulated masonry and brick construction.
3. Constant indoor and outdoor surface heat transfer coefficients prevail.
4. The thermal properties of conductivity and specific heat of the wall structure, and the density, are independent of temperature variations.
5. The variations of outdoor air temperature and solar intensity with time are identical on successive days.

Certain incidental assumptions are also employed to further simplify the statement of the method and the explanations of the analysis; these are:

1. The air inside the room is maintained at a constant temperature. (An assumption of periodic variation of inside temperature will not change the method).
2. The absorptivity of the outside surface to solar radiation is independent of the angle of incidence (any assumption of variation of absorptivity with angle of incidence may be made without changing the method).

Outside Surface Phenomena

An outside wall surface exposed to the sun is subject to heating or cooling in the following ways:

1. By convection to or from the surrounding atmosphere.
2. By radiation from the sun.
3. By radiation to or from the sky or surrounding objects.

An illustration of the foregoing may be obtained by referring to Fig. 1. Radiant energy from the sun falling on the outside surface of the wall is partially reflected and partially absorbed. The reflected portion has no influence on the heat flow through the wall. The absorbed portion of the solar energy, however, results in a rise in the outside surface temperature of the wall, as noted previously. By virtue of this rise in temperature of the outside surface the temperature gradient through the wall is increased, resulting in a higher rate of heat flow into the room. Heat also flows in the opposite direction from the outside surface by convection to the atmosphere and by radiation to the sky and surrounding objects.

The intensity of the incident solar radiation per square foot of wall area depends on the orientation of the wall, the time of day, the time of year, the latitude and the degree of cloudiness, smokiness, etc. Fig. 2, which is drawn from data presented previously,[1] illustrates the manner in which the incident solar radiation varies with orientation and time of day. The fraction of the incident radiation which is absorbed by the wall surface (absorptivity of the surface to solar radiation) depends upon the nature and color of the surface, and may also depend upon the angle of incidence. Thus, the solar heat input to one square foot of the wall can be expressed simply as aI, where a is the absorptivity of the surface to solar radiation and I is the instantaneous value of the sun intensity on one square foot of the wall surface.

In addition to receiving radiation from the sun, a square foot of wall surface emits to surrounding solid objects a net quantity of low temperature radiation, which in Btu per hour is given by

$$R - R' = 0.174 \times 10^{-8}\ \xi''\ (T_1^4 - T_s^4) \quad \ldots\ldots (1)$$

where

ξ'' = emissivity of low temperature radiation for the particular geometric relation between the radiating and receiving surface.

T_1 = temperature of the wall surface, degrees Fahrenheit, absolute.

T_s = temperature of surrounding solid objects, degrees Fahrenheit, absolute.

Over the small temperature ranges encountered in the present study, $(R - R')$ can be approximated by

$$R - R' = \xi''\ F\ (t'_1 - t_s) \quad \ldots\ldots (2)$$

$$R - R' = h_r\ (t'_1 - t_s) \quad \ldots\ldots (3)$$

where t'_1 and t_s are temperatures of the wall surface and surrounding solid objects in degrees Fahrenheit, and h_r is a radiation heat transfer coefficient equal to the product of ξ'' and F. F is a factor dependent upon the temperature level of the radiation heat transfer process, and

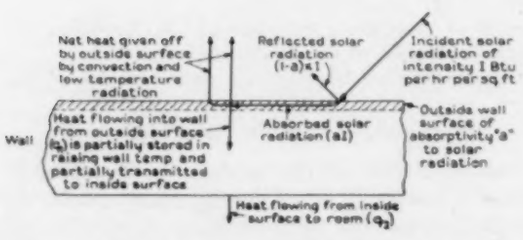
Net heat given off by outside surface by convection and low temperature radiation
Reflected solar radiation (1-a)I
Incident solar radiation of intensity I Btu per hr per sq ft
Heat flowing into wall from outside surface (q1) is partially stored in raising wall temp and partially transmitted to inside surface
Absorbed solar radiation (aI)
Outside wall surface of absorptivity "a" to solar radiation
Wall
Heat flowing from inside surface to room (q2)

Fig. 1—Heat flow phenomena in wall exposed to the sun

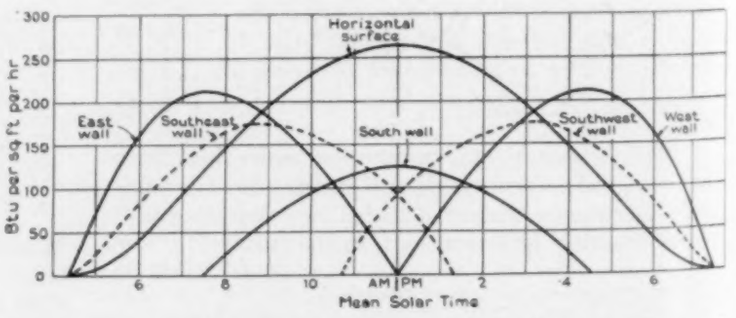

Fig. 2—Average midsummer intensity of solar radiation on various wall surfaces at 45 deg N latitude

is not the same as the F defined in a previous publication.[1] The possibility of expressing heat transfer by radiation in this manner when the temperature range is small has been pointed out by King.[5]

Horizontal surfaces, such as roofs, radiate to the sky an amount of heat, in Btu per hour per square foot,

$$R = 0.174 \times 10^{-8}\ \xi'\ T_1^4$$

where ξ' is the emissivity of the outside surface to low temperature radiation. However, the surface also receives an appreciable amount of radiant energy by radiation from water vapor in the lower regions of the atmosphere. For example, on a day when the air temperature at the surface is 70 F and a relative humidity of about 50 per cent prevails, approximately 105 Btu per hour per square foot is received by radiation from water vapor in the air (Appendix II). Since a black body at 70 F emits 137 Btu per hour per square foot, the net heat loss from a black horizontal surface at 70 F is approximately 32 Btu per hour per square foot.

For the sake of simplicity, the heat radiated and the heat absorbed by a surface facing the sky may be approximated, respectively, by

$$R = \xi' [R_b + F\ (t'_1 - t_b)] \quad \text{(4a)}$$
$$R' = \xi' [R_b' + F'\ (t_a - t_b)] \quad \text{(4b)}$$

where

R_b = heat radiated by a black body at a reference temperature t_b, Btu per hour per square foot.

R_b' = black-body absorption of radiation from an atmosphere whose temperature at the surface is t_b, Btu per hour per square foot.

t_a = temperature of surrounding air, degrees Fahrenheit.

F, F' = constants depending on the temperature level (F' is approximately 0.75 F.)

Convection heat transfer to a unit area of the outside surface is given by the familiar expression

$$h_1\ (t_a - t'_1) \quad \text{(5)}$$

where h_1 is the surface coefficient, dependent upon the wind velocity, and t_a and t'_1 are the instantaneous values of air temperature and outside surface temperature, respectively.

Air temperature as a function of the time of day varies so much from place to place that it is not possible to give a universal curve. It can be said, however, that it is nearly always a close approximation to an average value equal to the mean daily temperature plus an oscillating component with an amplitude equal to one half the *mean daily range*.[6] For most practical applications this can be approximated by a constant with a superposed sine wave having a 24-hour period. For more precise representation of the variation additional harmonics may be added to the expression by the method given in Appendix III. The maximum daily temperature usually occurs from one to three hours after noon, mean solar time (See Fig. 3).

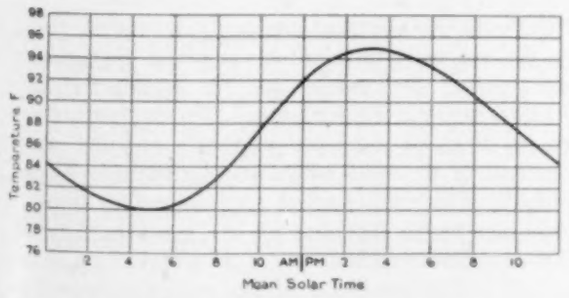

Fig. 3—**Approximation of typical variation in outdoor air temperature on a design day**

The components of heat transfer at the outside surface have now all been examined and they may be combined to give the equation for the heat balance at the outside surface, as follows:

$$q_1 = h_1\ (t_a - t'_1) + aI - (R - R') \quad \text{(6)}$$

where

q_1 = net heat flow from the outer surface to the interior of the wall, Btu per hour per square foot.

In most cases the assumption may be made, with sufficient accuracy, that the temperature of surrounding solid objects is equal to the air temperature. Hence, setting t_s equal to t_a, and substituting (2) in (6), the expression for walls facing terrestrial objects reduces to (7b), in which the radiation coefficient is lumped with the convection coefficient.

$$q_1 = (h_1 + \xi'' F)\ (t_a - t_s') + aI \quad \text{(7a)}$$
$$q_1 = H_1\ (t_a - t'_1) + aI \quad \text{(7b)}$$

For surfaces facing the sky the simplifying assumption is made that F and F' in equations (4) are equal. Substituting in (6), the expression for surfaces facing the sky becomes

$$q_1 = (h_1 + \xi' F)\ (t_a - t'_1) + aI - \xi'\ (R_b - R_b') \quad \text{(8a)}$$
$$q_1 = H_1'\ (t_a - t'_1) + (aI - \xi' \triangle R_b) \quad \text{(8b)}$$

Here a part of the low-temperature radiation effect is included in the outside film coefficient and the remainder is taken care of by the constant term ($\xi' \triangle R_b$).

Both equations (7) and (8) contain a term which is dependent on temperature difference and another term which is independent of temperature difference but dependent on intensity of radiant energy. Each one of these terms may vary in magnitude from hour to hour, as explained previously. Hence, a single solution to the differential equation written for the flow of heat through the wall is applicable to surfaces facing either the sky or terrestrial objects. Most practical applications may be reduced to either one or the other of these two cases. However, whenever the problem warrants more detailed study it is possible to use a mean somewhere between the two limiting cases given.

Conditions Within the Wall

An elementary thickness of wall just beneath the outside surface receives the net heat given to the wall by convection and radiation. A portion of this energy goes to heat up the element in question while the balance is conducted on to the next elementary thickness of the wall. Due to the fact that a certain time is required for the incident heat to raise the temperature of the element in question and pass on the remainder to the next element, there is a time lag between the input of a heat

[6] The *mean daily range* is the mean maximum daily temperature minus the mean minimum daily temperature.

impulse at the outer surface and its arrival at the inner surface. Furthermore, because the incident heat impulse has had some of its energy used in heating up the wall, the amplitude of the corresponding impulse is considerably reduced by the time it reaches the inner surface.

Thus, if the radiation falling on the outside surface of a wall were sinusoidal about a zero average value, a sinusoidal heat input to the room would be found from this source, lower in amplitude than that of the incident radiation and somewhat later in time phase. Similarly, if the outdoor temperature varied sinusoidally about the same temperature as that of the indoors, a sinusoidal heat flow at the inner surface would be found lower in amplitude and later in phase than that induced across the outside surface by the variation in air temperature. Actually, of course, the net radiation on the outside surface is not zero but some positive value; likewise, the mean outdoor temperature differs from that maintained indoors by some constant amount. These constants give rise to a constant heat flow through the wall which, superimposed upon the oscillating components at the inner surface, gives the total heat flow into the room.

Expression for Heat Flow

Having gained a physical picture of what occurs in the wall, a brief outline of the method of solution for the heat flow into the room will now be given. All temperatures are measured with respect to an assumed constant indoor air temperature for simplicity of mathematical expression. No generality is lost by this assumption, however, and the final solution is expressed in terms of the difference between indoor and outdoor temperatures.

The differential equation holding at all points in the wall is

$$k \frac{\partial^2 t}{\partial x^2} = \rho c \frac{\partial t}{\partial \theta} \quad \ldots\ldots (9)$$

which states that the difference between the heat flow in at one side of an elementary thickness ∂x and that out at the other side causes a local rate of rise in temperature dependent upon the heat capacity, ρc.

This must be solved subject to the following boundary condition at the outside surface, which is simply a restatement of equation (8b):

$$-k \left(\frac{\partial t}{\partial x}\right)_s = H_1' (t_s - t_1') + aI - \xi' \triangle R_b \quad \ldots\ldots (10)$$

The term on the left side, being the product of the gradient just inside the outer surface times the conductivity, represents the heat input to the wall.

The boundary condition at the inside surface is

$$-k \left(\frac{\partial t}{\partial x}\right)_2 = h_i t_2 \quad \ldots\ldots (11)$$

which states that the heat flow through the innermost element of the wall equals that transferred by convection (and radiation) to the room.

The solution of the differential equation (9), subject to boundary conditions (10) and (11), is:

$$q_2 = Ut_m + [V_o t_o \, Cos(\omega_o \theta - a_o - \emptyset_o) + V_1 t_1 \, Cos \, (\omega_1 \theta - a_1 - \emptyset_1) + \ldots]$$
$$+ \frac{U}{H_1'}(aI_m - \xi' \triangle R_b) + \frac{a}{H_1'} [V_o I_o \, Cos \, (\omega_o \theta - \beta_o - \emptyset_o) + V_1 I_1 Cos \, (\omega_1 \theta - \beta_1 - \emptyset_1) + \ldots] \quad \ldots\ldots (12)$$

This solution is derived in Appendix I and is based on the assumption that the temperature difference from outdoors to indoors and the intensity of incident solar heat can each be expressed as a constant plus a series of sinusoidal harmonics, as follows:

$$t = t_m + t_o Cos \, (\omega_o \theta - a_o) + t_1 Cos \, (\omega_1 \theta - a_1) + \ldots\ldots (13)$$

$$I = I_m + I_o \, Cos \, (\omega_o \theta - \beta_o) + I_1 \, Cos \, (\omega_1 \theta - \beta_1) + \ldots\ldots (14)$$

The symbols used in the expressions are defined at the end of the paper.

In equation (12) the term Ut_m represents the steady heat input to the room resulting from the excess of mean outdoor temperature over indoor temperature. This is the same as the well known expression for steady-state heat transfer by temperature difference used in all present standard methods of heat loss and heat gain calculations excepting that the temperature difference, t_m, is the mean temperature difference over a 24-hour period rather than the instantaneous difference between outdoor and indoor temperatures.

The term $\frac{U}{H_1'}(aI_m - \xi' \triangle R_b)$ is the heat input from mean solar radiation and net radiation to the sky. This term is equivalent to the expression for additional conduction through walls exposed to the sun given in a previous publication[1] excepting that the radiation intensity I_m is the average intensity over a 24-hour period rather than the instantaneous intensity, and excepting for the inclusion of a correction for net radiation to the sky.

Drawing an analogy with the electrical circuit, the two terms given may be likened to the direct current flowing when a constant voltage is applied, that is, $(1/U)$ is equivalent to the resistance of the circuit to direct current.

The first bracketed term in equation (12) is the oscillating component resulting from variation of outdoor temperature and the second bracketed term is the oscillating component resulting from variation in solar intensity. Referring again to the electrical analogy, these terms correspond with the current flowing in an electrical circuit having both a resistance and reactance to alternating voltages. That is, $(1/V_o, 1/V_1$, etc.) correspond to the impedances of the electrical circuit for alternating currents of various frequencies.

Drawing the electrical analogy still further, the angles $\emptyset_o$, $\emptyset_1$, etc. are equivalent to the phase displacement between the applied voltage and the resultant current.

The terms V_o, V_1, etc. and $\emptyset_o$, $\emptyset_1$, etc. in equation (12) are functions of the wall properties and the particular harmonic of air temperature or solar intensity being considered. The mathematical expressions for these terms, which are rather complicated, are given in Appendix I. However, the expressions for V and $\emptyset$ can be plotted against simple functions of the wall properties, as explained, thereby avoiding complicated numerical work in the application of the method.

In order to visualize the effect of the heat storage capacity of the wall upon the magnitude of the transmitted heat impulse, V_0, V_1, etc. may be divided by U. Thus, if

$$\lambda = \frac{V}{U} \quad \ldots\ldots (15)$$

equation (12) may be re-written as follows:

$$q_2 = U t_m + U [\lambda_0 t_0 Cos (\omega_0\theta - a_0 - \emptyset_0) + \lambda_1 t_1 Cos (\omega_1\theta - a_1 - \emptyset_1) + \ldots] + \frac{U}{H_1'}(a I_m - \xi' \triangle R_b) + \frac{U}{H_1'} a [\lambda_0 I_0 Cos (\omega_0\theta - \beta_0 - \emptyset_0) + \lambda_1 I_1 Cos (\omega_1\theta - \beta_1 - \emptyset_1) + \ldots] \ldots (16)$$

The solution given has been written for the case of a horizontal surface facing the sky, which gives the more general expression for the constant term in the radiation component. However, the solution for vertical walls facing terrestrial objects is identical, excepting that the term

$$\frac{U}{H_1'}(a I_m - \xi' \triangle R_b) \text{ becomes } \ldots \frac{U}{H_1}(a I_m)$$

Furthermore, for practical purposes it is reasonable to make the assumption that H'_1 and H_1 are equal for the following reasons:

1. For low temperature radiation the emissivity of the outer surface of most wall constructions, of surrounding solid objects and of the sky may be assumed nearly equal to unity This makes it possible to neglect the configuration factor in determining the effective emissivity of the outer wall surface.
2. The radiation component, h_r, is from 25 to 50 per cent of the outside film coefficient H_1. Hence, any error in the combined film coefficient is not over one-half the error in the assumed radiation coefficient, and is usually less.
3. The percentage error in the calculated heat flow is less than the percentage error in the assumed film coefficient.

The factor λ in (16) is the ratio of the rate of heat transmission through the wall for a sinusoidally variable temperature difference to the rate of heat transmission that would exist if the same temperature difference were maintained constant for an indefinite period of time. It is, therefore, a decrement factor. It is instructive to note that the same decrement factor λ and the same lag angle $\emptyset$ occur in both the sun effect term and in the term resulting from outdoor temperature variation. In other words, both the decrement factor and the lag angle are functions of the wall properties and are independent of the means by which the heat is received.

Under certain circumstances, harmonics of the temperature and solar intensity waves may be neglected. Not only are the amplitudes of the impulses smaller than that of the fundamental, but the effect felt at the inner surface is smaller because of a lower value of λ for the harmonics. This will be demonstrated later. In any case, it is very seldom that harmonics higher than the first must be considered.

Table 1—Vales of ω and Relative Values of σk for Different Sinusoidal Harmonics

Harmonic	Frequency, Cycles per Day	Period, Hours per Cycle	ω	Relative Value of (σk)
Fundamental	1	24	0.2618	1.00
1st harmonic	2	12	0.5236	1.41
2nd harmonic	3	8	0.7854	1.73
3rd harmonic	4	6	1.0472	2.00

Figs. 4 and 5 give charts of λ and $\emptyset$ vs. the wall characteristics and the frequency of the variation in temperature or solar intensity. The variables against which λ and $\emptyset$ are plotted in Figs. 4 and 5 are σk and L/k, where

$$\sigma = \sqrt{\frac{\rho c \omega}{2k}} \quad \ldots\ldots (17)$$

ρ = density of wall structure, pounds per cubic foot.
c = specific heat of wall structure, Btu per pound per degree Fahrenheit.
k = conductivity of wall structure, Btu per hour per square foot per degree Fahrenheit per foot.
L = thickness of wall, feet.
ω = angular velocity of sinusoidal variation, radians per hour.

It can be seen that L/k is constant for any given wall, but that σk depends on whether the fundamental or one of the harmonics of the variation is being considered. Table 1 gives values of ω and the relative magnitudes of σk for several harmonics.

The use of Figs. 4 and 5 will now be explained by means of an example. In order to preserve simplicity, it will be assumed that the wall is subjected to a variation in temperature, but that it is not exposed to the sun. The assumed variation in temperature difference between indoors and outdoors is

$$t = 10 + 15 Cos (15\theta - 210) + 2 Cos (30\theta - 240)$$

This corresponds to a maximum temperature difference of 23 F at 2 P. M. The assumed wall characteristics are:

$L/k = 7.0$
$\sigma k = 0.3$ (for the fundamental)
$U = 0.128$

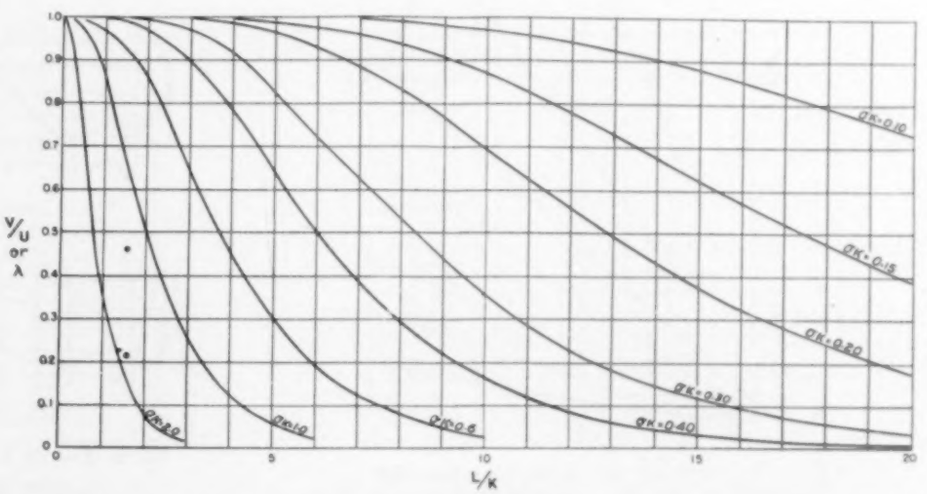

Fig. 4—Decrement factors for sinusoidal components of heat flow through homogeneous walls

Referring to Table 1, σk for the first harmonic can be determined, and then from Figs. 4 and 5 λ and ϕ can be found for the fundamental and the first harmonic. Table 2 summarizes the numerical results and also shows the magnitude of the mean, the fundamental and the first harmonic of the heat flow through the inside wall.

Neglecting the contribution of the first harmonic to the total heat flow (resulting in an error of not more than 3 per cent), the maximum flow of heat through the inside surface of the wall is 2.47 Btu per hour per square foot. With the same overall coefficient of heat transfer in a wall with zero heat storage capacity and with the same temperature difference, the maximum flow of heat would be

$0.128 \times 23 = 2.94$ Btu per hour.

To summarize, the effect of heat storage capacity in this example is to reduce the maximum rate of heat flow by 16 per cent and to delay the time when the maximum flow across the interior surface occurs by approximately 5.8 hours.

Correlation with Test Data

An interesting check on the method which has been presented is obtained by comparing calculated heat flows at the inner surface with those obtained in tests by Houghten et al.[3] The following examples are for two different panels, one tested on July 27, 1931 and the other on September 10, 1931. These are the only two days on which the reported test data include measurements of both outdoor air temperature and intensity of solar radiation.

The first case is that of a 2-in. thick horizontal pine panel tested on July 27, 1931. The solar intensity on a horizontal surface for that day is shown in Fig. 6a.

Table 2—Summary of Illustrative Example

Component	L/k	σk	λ (Fig. 4)	t (Max. Value)	$q = \lambda U t$	ϕ (Fig. 5) Deg	ϕ (Fig. 5) Hours
Mean	7.0	—	—	10	1.28	—	—
Fundamental	7.0	0.30	0.62	15	1.19	87	5.8
1st Harmonic	7.0	0.42	0.33	2	0.08	140	4.7

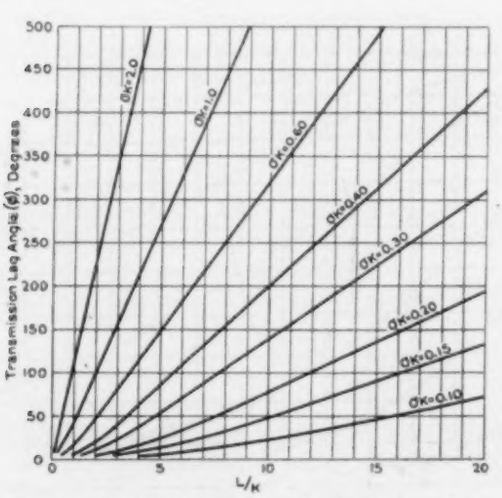

Fig. 5—Transmission lag angles for sinusoidal components of heat flow through homogeneous walls

When analyzed by the method given in Appendix III this may be expressed as

$$I = 58 - 70.5 \cos 15\theta - 8.3 \sin 15\theta + 38.5 \cos 30\theta + 3.3 \sin 30\theta + \ldots = 58 + 71.0 \cos (15\theta - 187) + 38.6 \cos (30\theta - 5) + \ldots \quad (18)$$

The outdoor air temperature for the same day is shown in Fig. 6b. In a similar manner it is expressed as

$$t = 9.4 - 13.9 \cos 15\theta - 8.8 \sin 15\theta + 1.7 \cos 30\theta - 1.0 \sin 30\theta + \ldots = 9.4 + 16.4 \cos (15\theta - 212) + 2.0 \cos (30\theta - 329) + \ldots \quad (19)$$

where the indoor temperature of 69.6 F is the zero reference.

Using the following numerical values given in the previous paper,[3] the heat flow at the inner surface was calculated through the use of equation (12) and the expressions for V and ϕ given in detail in Appendix I.

$H_1 = 2.0$ Btu per hour — ft² —F
$h_2 = 1.9$ Btu per hour — ft² —F
$k = 0.069$ Btu per hour — ft —F
$c = 0.467$ Btu per pound —F
$\rho = 37.5$ lb per cubic foot
$L = 0.180$ ft
$\Delta R_b = 32$ Btu per hour—ft² (supplied by authors)
$a = \xi' = 1.0$ (supplied by authors)

The assumption of unity for absorptivity to solar radiation and emissivity to low temperature radiation is based on the explanation[3] that the outside surfaces of the panels were painted with lampblack pigment.

The calculated results are plotted as the broken line in Fig. 6c. The solid line in the same figure indicates

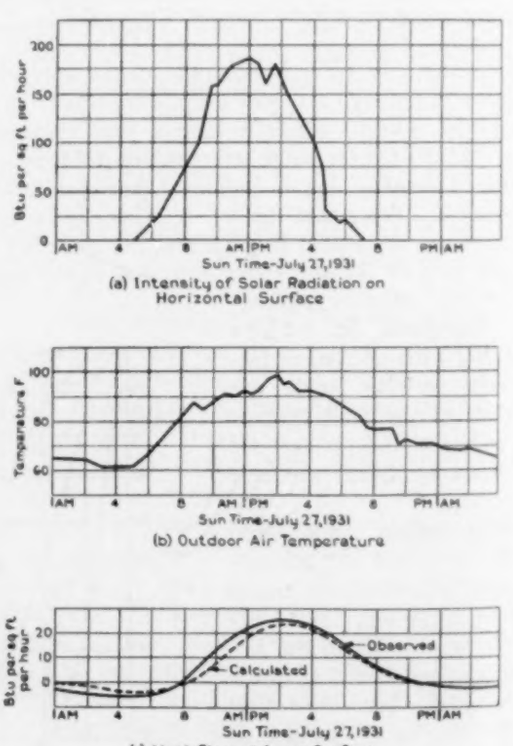

Fig. 6—Test on 2-in. pine panel, July 27, 1931
(a) Solar intensity on horizontal surface
(b) Outdoor air temperature
(c) Heat flow into room

the test results. The agreement is well within limits with which atmospheric conditions could be predicted for design purposes.

The second case is that of a 4-in. thick gypsum panel, similarly oriented, and tested on September 10, 1931. The curve of solar intensity for that day is given in Fig. 7a, and may be mathematically expressed as

$$I = 37 - 59.6\cos 15\theta + 2.3\sin 15\theta + 27.0\cos 30\theta - 1.9\sin 30\theta + \ldots = 37 + 59.6\cos(15\theta - 178) + 27.4\cos(30\theta - 356) + \ldots \quad (20)$$

Outdoor air temperature for the day is shown in Fig. 7b and may be expressed as

$$t = 10.4 - 13.3\cos 15\theta - 7.4\sin 15\theta + 1.6\cos 30\theta + 1.1\sin 30\theta + \ldots = 10.4 + 15.2\cos(15\theta - 209) + 2.0\cos(30\theta - 35) + \ldots \quad (21)$$

where the indoor temperature of 69.6 F is the zero reference.

Constants used in calculating the heat flow at the inner surface as before are:

$h_1 = 2.0$ Btu per hour — ft² — F
$h_2 = 1.9$ Btu per hour — ft² — F
$k = 0.12$ Btu per hour — ft — F
$c = 0.234$ Btu per pound — F
$\rho = 64.9$ lb per ft³
$L = 0.350$ ft
$R_b = 32$ Btu per hour — ft² (supplied by authors)
$a = \xi' = 1.0$ (supplied by authors)

The results are plotted as the broken line in Fig. 7c. Again a satisfactory agreement with test results is obtained.

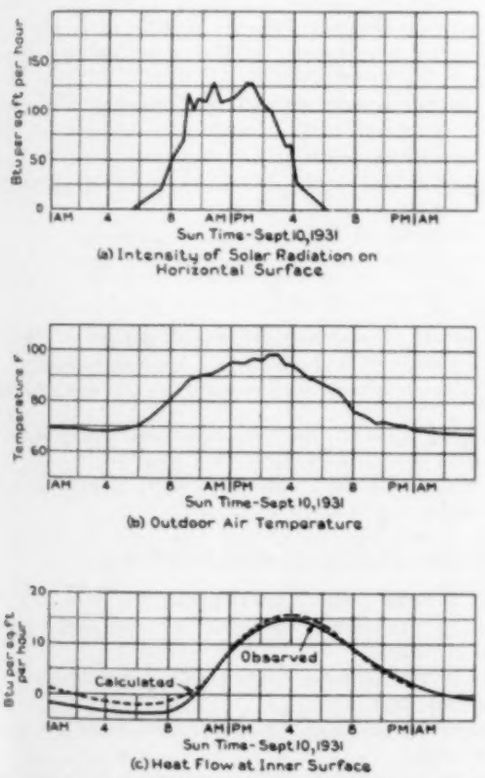

Fig. 7—Test on 4-in. gypsum panel, September 10, 1931

(a) Solar intensity on horizontal surface
(b) Outdoor air temperature
(c) Heat flow into room

Sample Calculation

A complete numerical example is given to illustrate the application of the method which has been given to a homogeneous wall exposed to the sun.

Consider an 8-in. brick wall with an unobstructed southern exposure at latitude 45 deg. Assume that for design conditions the solar intensity as given by Fig. 2 may be used, and that the outdoor air temperature given for July 27, 1931, at Pittsburgh, Pa.[3] may be used. Assume, also, that the design indoor temperature is 80 F. The equations for solar intensity and temperature difference, outdoors-to-indoors, may be derived by the method given in Appendix III.

This results in the following, where all harmonics above the first are neglected:

$$I = 29.4 + 52.4\,Cos\,(15\theta - 180) + 34.8\cos 30\theta \quad (22)$$

$$t_o = 79.0 + 16.4\,Cos\,(15\theta - 212) + 2.0\cos(30\theta - 329) \quad (23)$$

$$t = -1 + 16.4\,Cos\,(15\theta - 212) + 2.0\cos(30\theta - 329) \quad (24)$$

where

t_o = outdoor air temperature
t = difference between outdoor and indoor air temperatures

The wall constants are now evaluated. The following quantities are obtained from the references indicated:

$k = 0.42$ Btu per hour — ft — F ([4])
$\rho = 132$ lb per ft³ ([7])
$c = 0.22$ Btu per pound — F ([7])
$h_2 = 1.65$ Btu per hour — ft² — F ([5])
$H_1 = 4.3$ Btu per hour — ft² — F (Supplied by authors)[b]
$a = 0.7$ ([1])

For the fundamental wave, $\omega_0 = 2\pi/24 = 0.262$ radians per hour or 15 deg per hour. For the first harmonic, $\omega_1 = 2\pi/12 = 0.524$ radians per hour or 30 deg per hour. From the above,

$$\sigma_0 k = \sqrt{\frac{\rho c k \omega_0}{2}} = \sqrt{\frac{(132)\,(0.22)\,(0.42)\,(0.262)}{2}} = 1.27$$

and

$$\sigma_1 k = \sqrt{\frac{\rho c k \omega_1}{2}} = \sqrt{\frac{(132)\,(0.22)\,(0.42)\,(0.524)}{2}} = 1.80$$

The factor $L/k = \dfrac{(8)}{(12)\,(0.42)} = 1.58$

The overall heat transfer coefficient of the wall is

$$U = \frac{1}{\frac{1}{H_1} + \frac{1}{h_2} + \frac{L}{k}} = \frac{1}{\frac{1}{4.3} + \frac{1}{1.65} + 1.58} = 0.41$$

Referring to Figs. 4 and 5, the following values of λ and $\emptyset$ are found for the previous values of σk and L/k:

$\lambda_0 = 0.46$ $\quad$ $\emptyset_0 = 106$
$\lambda_1 = 0.21$ $\quad$ $\emptyset_1 = 163$

Finally, it is assumed that the wall faces either ter-

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[b] This numerical value, and Figs. 4 and 5 are based on surface coefficients, as follows:

$H_1 = 4.3$ (supplied by authors)
$h_2 = 1.65$ ([5])

The outside surface coefficient is based on a wind velocity of 8.1 mph, which is the average summer velocity for 69 cities listed in the 1938 Guide.[6] It is derived from the standard value of 6.0[6] based on 15 mph wind velocity and used for winter design calculations, as follows:

From King,[5] the effective radiation coefficient for black bodies at ordinary temperature ranges is 1.3. From the same source, the convection coefficient varies as the 0.75 power of velocity. Hence

$$(H_1)_{0\text{-}1\text{mph}} = 1.3 + (6.0 - 1.3)\left(\frac{8.1}{15}\right)^{0.75} = 4.26$$

The assumption that outside surfaces of average wall structures are very nearly black bodies to low temperature radiation is sufficiently correct for practical purposes. The error involved is small, as an assumption of $\epsilon = 0.75$ results in a combined outside coefficient of 4.14 in place of 4.26

restrial objects of unit emissivity and atmospheric temperature, or layers of air near the earth of sufficient thickness to be effectively a black body. Here $\triangle R_b = 0$.

All constants required in equation (16) have now been determined. Substituting these constants, the equation for heat flow through one square foot of the inner surface of the wall in question is:

$$q_2 = (0.41)(-1.0) + (0.41)\left[\begin{matrix}(0.46)(16.4)\ Cos\ (15\theta - 212 - 106) \\ + (0.21)\ (2.0)\ Cos\ (30\theta - 329 - 163)\end{matrix}\right] + \frac{(0.41)}{(4.3)}(0.7)(29.4) + \frac{(0.41)}{(4.3)}(0.7)\left[\begin{matrix}(0.46)(52.4)\ Cos\ (15\theta - 180 - 106) \\ + (0.21)(34.8)\ Cos\ (30\theta - 163)\end{matrix}\right]$$

$$q_2 = 1.53 + 3.10\ Cos\ (15\theta - 318) + 0.17\ Cos\ (30\theta - 132) + 1.61\ Cos\ (15\theta - 286) + 0.49\ Cos\ (30\theta - 163) \quad \ldots\ldots\ldots (25)$$

Equation (25) may be plotted easily over whatever period is of interest. Fig. 8 gives it in its entirety. Fig. 8 also shows, for purposes of comparison, what the calculated heat flow would be if heat storage capacity were neglected. This is calculated from Equation (26) which is the same as (25), excepting that $\lambda_0 = \lambda_1 = 1.0$ and $\emptyset_0 = \emptyset_1 = 0$.

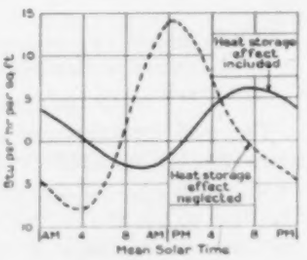

Fig. 8—Effect of heat storage on heat transmission through an 8-in. brick wall

$$q_2 = 1.55 + 6.72\ Cos\ (15\theta - 212) + 0.82\ Cos\ (30\theta - 329) + 3.50\ Cos\ (15\theta - 180) + 2.32\ Cos\ 30\theta \quad \ldots\ldots\ldots\ldots\ldots (26)$$

Conclusions

1. This paper has presented a rational method for calculating the heat flow through a homogeneous wall that experiences a diurnal variation of air temperature and solar intensity on its outside surface, and is bordered at the inside by room air at constant temperature. The method avoids a determination or estimate of the outside surface temperature of the wall, and gives the heat flow into the room, as a function of time, in terms of the wall constants and the design conditions of solar intensity and outdoor air temperature. The analysis shows that a heat impulse of a given amplitude entering the wall at the outer surface requires an appreciable time to pass through, and that the amplitude of the heat impulse is reduced. The amount of time lag and the reduction in amplitude depend upon the thickness and thermal properties of the wall, and can be determined easily from charts.

2. It is believed that the method presented here is sufficiently simple so that it can be used in practical applications where certain design outdoor and indoor conditions are assumed. However, some further simplifications can undoubtedly be made to permit consideration of the influence of the heat storage effect with a minimum of time and effort. Tabulations of wall properties not easily found at present in standard handbooks and more extensive charts than those presented here will be of material benefit.

3. The method presented in this paper does not solve the problem for non-homogeneous walls, which is much more complicated than that for homogeneous walls. However, the authors are of the opinion that further analysis and further comparison with test results on various types of wall construction will permit the development of a single method which may be applied with practical accuracy to any type of wall structure; that further study will lead to the development of reasonably simple methods which can be used to determine constants for various wall structures to be used in such a method.

Notation

Letter symbols used in this paper are defined as follows:

- a = absorptivity of outer wall surface to solar radiation.
- c = specific heat of wall structure, Btu per pound — deg F.
- F = effective radiation heat transfer coefficient for black body, Btu per hour — ft² — deg F.
- F' = effective radiation heat transfer coefficient for radiation from the atmosphere, Btu per hour — ft² — deg F.
- h = convection surface heat transfer coefficient, Btu per hour —ft² — deg F.
- H = combined convection and radiation surface heat transfer coefficient, Btu per hour — ft² — deg F.
- I = intensity of solar radiation on wall surface, Btu per hour — ft².
- k = thermal conductivity of wall structure, Btu per hour — ft — deg F.
- L = total wall thickness, feet.
- q = heat flow, Btu per hour — ft².
- R = outgoing low temperature radiation, from wall, Btu per hour — ft².
- R' = low temperature radiation received by wall, Btu per hour — ft².
- R_b = value of R at a reference temperature, Btu per hour — ft².
- R_b' = value of R' at a reference temperature, Btu per hour — ft².
- $\triangle R_b = R_b - R_b'$.
- t = temperature, degrees Fahrenheit, usually referred to indoor temperature, for zero.
- T = temperature, degrees Fahrenheit, absolute.
- U = overall heat transfer coefficient of wall to constant temperature difference, Btu per hour — ft² — deg F.
- V = overall heat transfer coefficient of wall to sinusoidally varying temperature.
- x = distance through the wall, feet, measured from the center.
- α = displacement angle of outdoor air temperature wave referred to midnight as zero, radians.
- β = displacement angle of solar intensity wave referred to midnight as zero, radians.
- ξ = emissivity to low temperature radiation.
- λ = decrement factor for amplitude of transmitted heat impulse.
- $\lambda = V/U$.
- ρ = density of wall structure, pounds per ft³.
- $\emptyset$ = lag angle of transmitted heat impulse, radians.
- ω = angular velocity of sinusoidal waves, radians per hour.
- $\sigma = \sqrt{\frac{\rho c \omega}{2k}}$
- θ = mean solar time, number of hours measured from midnight.

Subscript symbols are used as follows.

- m refers to mean value of heat impulse.
- o refers to fundamental component of sinusoidal variation (1 cycle per day).
- 1 refers to first harmonic component of sinusoidal variation (2 cycles per day).
- i refers to conditions at the outside surface of the wall in all other cases.
- 2 refers to second harmonic component of sinusoidal variation (3 cycles per day).
- 2 refers to conditions at the inside surface of the wall in all other cases.
- s refers to surrounding terrestrial objects.

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Appendix I—Mathematical Solution of the Heat Flow Equation

This Appendix gives the derivation for the heat flow through the inner surface of a homogeneous wall which is exposed on the outside to a variable air temperature and a variable intensity of solar radiation. The analysis employs the method of the complex variable.

In addition to the letter symbols given in the main body of the paper, the following are used in this Appendix:

$d = \frac{L}{2}$ = one half the wall thickness

e = base of natural logarithms

$m = \sqrt{j} = \sqrt[4]{-1}$

$j = \sqrt{-1}$

f, g, V, Z = functions of wall properties

A, B, C, D, p = constants

$$\sigma = \sqrt{\frac{\rho c \omega}{2k}}$$

A section of the wall is shown in Fig. I-1. For convenience, x is measured from the midpoint. The general differential equation for unidirectional heat flow in the x direction is

$$\frac{\partial^2 t}{\partial x^2} = \frac{\rho c}{k}\frac{\partial t}{\partial \theta} \quad \ldots\ldots (I\text{-}1)$$

For the sake of simplicity, only the constant term and one oscillating component term (the fundamental) of t and I will be used in the solution which follows. At the end, the solution for any harmonic may be written at once from the solution for the fundamental. This may then be superposed upon the basic solution to obtain the more complete and accurate one.

This must be solved subject to the following boundary conditions:

(1) At the inside surface $(x = d)$

$$-k\left(\frac{\partial t}{\partial x}\right)_1 = h_2\, t_2 \quad \ldots\ldots (I\text{-}2)$$

where t_2 is the temperature of the inside wall surface referred to room air as zero.

(2) At the outside surface $(x = -d)$

$$-k\left(\frac{\partial t}{\partial x}\right)_1 = h_1 (t_a - t_1') + aI - R + R' \quad \ldots (I\text{-}3)$$

where t_1' and t_a are temperatures of the outer wall surface and of outside air, respectively.

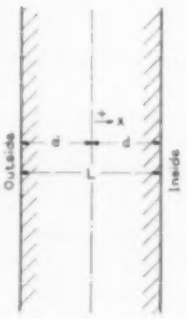

Fig. I-1—Cross section of wall

As an approximation, R and R' can be represented, respectively, by

$$R = \xi' [R_b + F (t_1' - t_b)] \quad \ldots\ldots (I\text{-}4a)$$

$$R' = \xi' [R_b' + F' (t_a - t_b)] \quad \ldots\ldots (I\text{-}4b)$$

where R_b and R'_b are values of low temperature radiation from the outer wall surface and from the atmosphere, respectively, at a reference temperature t_b.

In Appendix II it is shown that $F' \cong 0.8\ F$, so that it is fair to assume, at least for values of t_1' and t_a not too far removed from t_b in absolute temperature, that $F = F'$. Then (I-3) becomes

$$\begin{aligned} -k\left(\frac{\partial t}{\partial x}\right)_1 &= (h_1 + \xi' F)(t_a - t_1') + aI - \xi'(R_b - R_b') \\ &= H_1 (t_a - t_1') + aI - \xi'(R_b - R_b') \\ &= H_1 \{ [t_m + t_o \cos(\omega_o\theta - \alpha_o) + t_1 \cos(\omega_1\theta - \alpha_1) \\ &\quad + \ldots] - t_1' \} \\ &\quad + a \{ I_m + I_o \cos(\omega_o\theta - \beta_o) + I_1 \cos(\omega_1\theta - \beta_1) \ldots \} \\ &\quad - \xi' \triangle R_b \quad \ldots\ldots (I\text{-}3a) \end{aligned}$$

where t_m, t_o, t_1, etc., refer to outdoor air temperature and t_1' refers to the temperature of the outer wall surface, both referred to inside or room air as zero.

Assume a solution to (I-1) of the form

$$t = A + Bx + e^{-m^2\omega_o\theta}(C \cos p_o mx + D \sin p_o mx) \quad \ldots\ldots (I\text{-}5)$$

in which A, B, C and D may be either real or complex numbers. Let it be understood that the real part of the resulting complex value of t will constitute the actual physical solution.

Substituting (I-5) into (I-1), there results

$$\begin{aligned} & e^{-m^2\omega_o\theta}(-Cp_o^2m^2 \cos p_o mx - Dp_o^2m^2 \sin p_o mx) \\ &= -\frac{\rho c m^2 \omega_o}{k} e^{-m^2\omega_o\theta}(C \cos p_o mx + D \sin p_o mx) \quad \ldots\ldots (I\text{-}6) \end{aligned}$$

Since this must hold for all values of θ and x, it is an identity and the coefficients of corresponding terms can be equated to obtain

$$p_o = \sqrt{\frac{\rho c \omega_o}{k}} \quad \ldots\ldots (I\text{-}7)$$

With this value for p_o determined, (I-5) has been made to satisfy the differential equation (I-1).

In order to evaluate A, B, C and D, (I-5) is substituted into the boundary condition equations (I-2) and (I-3), reduced to complex form. Substituting (I-5) into the complex equivalent of (I-3a)

$$\begin{aligned} & -B - e^{-m^2\omega_o\theta}\ [Cp_o m \sin p_o md + Dp_o m \cos p_o md] \\ &= \frac{H_1 t_m + aI_m - \xi' \triangle R_b}{k} + e^{-m^2\omega_o\theta}\left[\frac{(H_1 t_o e^{m^2\alpha_o} + aI_o e^{m^2\beta_o})}{k}\right] \\ & -\frac{H_1}{k}\left[A - Bd + e^{-m^2\omega_o\theta}(C \cos p_o md - D \sin p_o md)\right] \quad (I\text{-}8) \end{aligned}$$

Substituting (I-5) into (I-2)

$$\begin{aligned} & -B - e^{-m^2\omega_o\theta}\left[-Cp_o m \sin p_o md + Dp_o m \cos p_o md\right] = \\ & \frac{h_2}{k}\left[A + Bd + e^{-m^2\omega_o\theta}(C \cos p_o md + D \sin p_o md)\right] \quad \ldots (I\text{-}9) \end{aligned}$$

The boundary conditions must, of course, hold true at all values of θ; hence (I-8) and (I-9) are identities in which the coefficients of the $e^{-m^2\omega_o\theta}$ terms on either side may be equated. For the same reason, the constants on either side may be equated. Writing equations of the constants and coefficients of (I-8) respectively,

$$-B=\frac{H_1t_m+aI_m-\xi'\triangle R_b}{k}-\frac{H_1}{k}(A-Bd) \quad \text{.....(I-10)}$$

and

$$Cp_om\sin p_omd+Dp_om\cos p_omd=-\frac{H_1}{k}(C\cos p_omd-D\sin p_omd)$$
$$+\frac{H_1t_o\,e^{m^2\alpha_o}+aI_o\,e^{m^2\beta_o}}{k} \quad \text{.....(I-11)}$$

Similarly, the equations of the constants and coefficients of (I-9) are, respectively,

$$-B=\frac{h_2}{k}(A+Bd) \quad \text{.....(I-12)}$$

$$Cp_om\sin p_omd-Dp_om\cos p_omd=\frac{h_2}{k}(C\cos p_omd+D\sin p_omd) \quad \text{(I-13)}$$

Solving (I-10) and (I-12) simultaneously,

$$A=\frac{(k+dh_2)\,(H_1t_m+aI_m-\xi'\triangle R_b)}{k(H_1+h_2)+2dH_1h_2} \quad \text{.....(I-14)}$$

and

$$B=-\frac{h_2(H_1t_m+aI_m-\xi'\triangle R_b)}{k(H_1+h_2)+2dH_1h_2} \quad \text{.....(I-15)}$$

Solving (I-11) and (I-13) simultaneously while making use of the fact that

$$m=\frac{1+m^2}{\sqrt{2}},\ \sin m^2pd=m^2\sinh pd \text{ and } \cos m^2pd=\cosh pd,$$

$$C=\frac{H_1t_o\,e^{m^2\alpha_o}+aI_o\,e^{m^2\beta_o}}{k(f_o+m^2g_o)}(p_om\cos p_omd+\frac{h_2}{k}\sin p_omd) \quad \text{(I-16)}$$

and

$$D=\frac{H_1t_o\,e^{m^2\alpha_o}+aI_o\,e^{m^2\beta_o}}{k(f_o+m^2g_o)}(p_om\cos p_omd+\frac{h_2}{k}\sin p_omd) \quad \text{(I-17)}$$

where

$$f_o=\frac{H_1h_2}{k^2}\sin\sqrt{2}\,p_od\cosh\sqrt{2}p_od+p_o^2\cos\sqrt{2}\,p_od\sinh\sqrt{2}\,p_od$$
$$+\frac{p_o(H_1+h_2)}{\sqrt{2}\,k}(\cos\sqrt{2}\,p_od\cosh\sqrt{2}\,p_od+\sin\sqrt{2}\,p_od\sinh\sqrt{2}\,p_od)$$
$$\text{.....(I-18)}$$

$$g_o=\frac{H_1h_2}{k_2}\cos\sqrt{2}\,p_od\sinh\sqrt{2}\,p_od-p_o^2\sin\sqrt{2}\,p_od\cosh\sqrt{2}\,p_od$$
$$+\frac{p_o(H_1+h_2)}{\sqrt{2}\,k}(\cos\sqrt{2}\,p_od\cosh\sqrt{2}\,p_od-\sin\sqrt{2}\,p_od\sinh\sqrt{2}p_od)$$
$$\text{.....(I-19)}$$

With the constants of (I-5) thus evaluated so as to satisfy the boundary conditions, it is possible to find t_2 which, when multiplied by h_2, gives the heat input to the room. Substituting (I-14), (I-15), (I-16) and (I-17) into (I-5) and evaluating it at $x=d$, after simplifying and rationalizing,

$$t_2=\frac{H_1t_m+aI_m-\xi'\triangle R_b}{H_1+h_2+\dfrac{H_1h_2L}{k}}$$
$$+\frac{H_1t_o}{k(f_o^2+g_o^2)}\left\{\left[\frac{f_o+g_o}{\sqrt{2}}\cos(\omega_o\theta-\alpha_o)+\frac{f_o-g_o}{\sqrt{2}}\sin(\omega_o\theta-\alpha_o)\right]\right.$$
$$\left.+j\left[\frac{f_o-g_o}{\sqrt{2}}\cos(\omega_o\theta-\alpha_o)-\frac{f_o+g_o}{\sqrt{2}}\sin(\omega_o\theta-\alpha_o)\right]\right\}$$
$$+\frac{aI_op}{k(f_o^2+g_o^2)}\left\{\left[\frac{f_o+g_o}{\sqrt{2}}\cos(\omega_o\theta-\beta_o)+\frac{f_o-g_o}{\sqrt{2}}\sin(\omega_o\theta-\beta_o)\right]\right.$$
$$\left.+j\left[\frac{f_o-g_o}{\sqrt{2}}\cos(\omega_o\theta-\beta_o)-\frac{f_o+g_o}{\sqrt{2}}\sin(\omega_o\theta-\beta_o)\right]\right\} \text{..(I-20)}$$

Since it was understood at the beginning that the real part of t would constitute the physical solution, the imaginary part of (I-20) can be discarded. Multiplying the remainder by h_2, the heat input to the room is

$$h_2t_2=\frac{h_2(H_1t_m+aI_m-\xi'\triangle R_b)}{H_1+h_2+\dfrac{H_1h_2L}{k}}+\frac{h_2H_1t_op}{k(f_o^2+g_o^2)}$$
$$\left[\frac{f_o+g_o}{\sqrt{2}}\cos(\omega_o\theta-\alpha_o)+\frac{f_o-g_o}{\sqrt{2}}\sin(\omega_o\theta-\alpha_o)\right]$$
$$+\frac{h_2aI_op}{k(f_o^2+g_o^2)}\left[\frac{f_o+g_o}{\sqrt{2}}\cos(\omega_o\theta-\beta_o)+\frac{f_o-g_o}{\sqrt{2}}\sin(\omega_o\theta-\beta_o)\right]$$
$$\text{.....(I-21)}$$

To make the numerical calculation easier, let

$$\sigma_o=\frac{p}{\sqrt{2}}=\sqrt{\frac{pc\omega_o}{2k}} \quad \text{.....(I-22)}$$

$$Y_o=\frac{f_o+g_o}{2\sigma_o^2}=(\frac{H_1h_2}{2\sigma_o^3k^2}+1)\cos\sigma_oL\sinh\sigma_oL+(\frac{H_1h_2}{2\sigma_o^3k^2}-1)$$
$$\sin\sigma_oL\cosh\sigma_oL+(\frac{H_1+h_2}{\sigma_ok})\cos\sigma_oL\cosh\sigma_oL \quad \text{.....(I-23)}$$

$$Z_o=(\frac{H_1h_2}{2\sigma_o^3k^2}+1)\sin\sigma_oL\cosh\sigma_oL-(\frac{H_1h_2}{2\sigma_o^3k^2}-1)$$
$$\cos\sigma_oL\sinh\sigma_oL+(\frac{H_1+h_2}{\sigma_ok})\sin\sigma_oL\sinh\sigma_oL \quad \text{.....(I-24)}$$

Finally, define an angle $\emptyset$ such that

$$\sin\emptyset_o=\frac{f_o-g_o}{\sqrt{2}\sqrt{f_o^2+g_o^2}}$$

$$\cos\emptyset_o=\frac{f_o+g_o}{\sqrt{2}\sqrt{f_o^2+g_o^2}} \quad \text{.....(I-25)}$$

from which

$$\emptyset_o=\tan^{-1}\frac{Z_o}{Y_o} \quad \text{.....(I-26)}$$

where $\sin\emptyset_o$ has the sign of Z_o
$\cos\emptyset_o$ has the sign of Y_o

Simplifying (I-21) with the aid of (I-23), (I-24), and (I-25), the solution for the constant term and the fundamental is:

$$q_2=h_2t_2=\frac{h_2(H_1t_m+aI_m-\xi'\triangle R_b}{H_1+h_2+\dfrac{H_1h_2L}{k}}$$
$$+\frac{h_2}{k\sigma_o\sqrt{Y_o^2+Z_o^2}}\left[H_1t_o\cos(\omega_o\theta-\alpha_o-\emptyset_o)\right.$$
$$\left.+aI_o\cos(\omega_o\theta-\beta_o-\emptyset_o)\right] \quad \text{.....(I-27)}$$

or

$$q_2=Ut_m+V_ot_o\cos(\omega_o\theta-\alpha_o-\emptyset_o)$$
$$+\frac{U(aI_m-\xi'\triangle R_b)}{H_1}+\frac{V_oaI_o}{H_1}\cos(\omega_o\theta-\beta_o-\emptyset_o) \quad \text{.....(I-28)}$$

where

$$U = \frac{1}{\frac{1}{H_1} + \frac{1}{h_2} + \frac{L}{k}} \quad \text{.....(I-29)}$$

$$V_o = \frac{H_1 h_2}{k\sigma_o \sqrt{Y_o^2 + Z_o^2}} \quad \text{.....(I-30)}$$

As stated earlier, the solution for the response to harmonics of the outdoor temperature and solar intensity wave will be of the same form as the second and fourth terms of (I-28) respectively, and may be superposed on the basic solution. The complete solution is then

$$q_i = Ut_m + [V_o t_o cos(\omega_o\theta - \alpha_o - \phi_o) + V_1 t_1 cos(\omega_1\theta - \alpha_1 - \phi_o) + \ldots] + \frac{U(aI_m - \xi' \triangle R_b)}{H_1} + \frac{a}{H_1}[V_o I_o cos(\omega_o\theta - \beta_o - \phi_o) + V_1 I_1 cos(\omega_1\theta - \beta_1 - \phi_1) + \ldots] \quad \text{.....(I-31)}$$

Finally, defining a decrement factor

$$\lambda = \frac{V}{U} \quad \text{.....(I-32)}$$

the solution can be written

$$q_i = Ut_m + U[\lambda_o I_o cos(\omega_o\theta - \alpha_o - \phi_o) + \lambda_1 t_1 cos(\omega_1\theta - \alpha_1 - \phi_1) + \ldots] + \frac{U}{H_1}(aI_m - \xi' \triangle R_b) + \frac{U}{H_1} a[\lambda_o I_o cos(\omega_o\theta - \beta_o - \phi_o) + \lambda_1 I_1 cos(\omega_1\theta - \alpha_1 - \phi_1) + \ldots] \quad \text{.....(I-33)}$$

Appendix II
Radiation of Atmospheric Water Vapor

The amount of radiation emitted by a layer of water vapor depends upon both its temperature and the product of the vapor pressure times the average distance from points in the vapor layer to the receiving surface.

Applying this to radiation from atmospheric water vapor, it is assumed that:

1. All radiation reaching the earth originates in the first 5000 ft of elevation.
2. The vapor pressure throughout this layer corresponds to that of 50 per cent relative humidity at the temperature of surface air.
3. The average temperature of origin for radiation reaching the surface from the atmosphere is 5 F below that of the surface air.

Assumption (1) is justified by the fact that radiation originating from more remote regions is practically all absorbed before reaching the surface.

Although the normal atmospheric temperature gradient is about 18 F per 5000 ft elevation, the larger proportion of radiation received from the warmer vapor layers near the surface makes the 5 deg average value of assumption (3) seem reasonable.

Turning to the chart for water vapor radiation given by Hottel and Mangelsdorf,[9] the radiation from water vapor in the atmosphere, R', is plotted against temperature for various values of the product of water vapor pressure in atmospheres, P_v, and average radiating distance in feet, X. At 65 F, R' is as shown in Fig. II-1.

In the present case, assuming a surface temperature of 70 F (at which temperaure the saturation pressure of water vapor is 0.3628 lb per square inch, or 18.76 mm of mercury) the vapor pressure at 50 per cent relative humidity is:

$$P_v = (0.50)\left(\frac{18.8}{760}\right) = 0.0123 \text{ atmospheres}$$

Based on the statement of Hottel in an earlier paper[5] that $X = 1.8$ times the layer thickness, for layers bounded by parallel planes,

$$X = (1.8)(5000) = 9000 \text{ ft.}$$

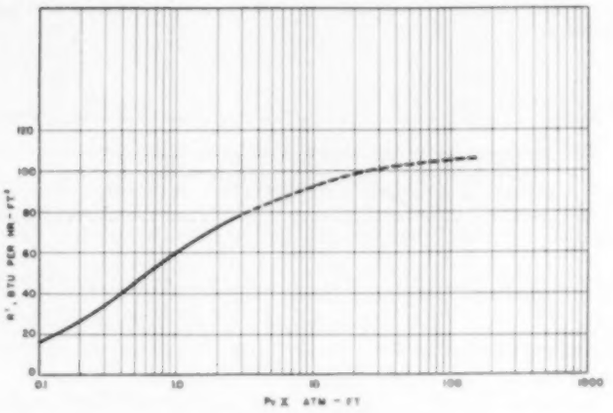

Fig. II-1—Radiation from water vapor at 65 F

Then

$$p_v X = (0.0123)(9000) = 112 \text{ atm.-ft.}$$

Referring to Fig. II-1 which has been mathematically extrapolated to large values of $p_v X$, it is found that R' for $p_v X = 112$ is equal to 105 Btu per hour-ft².

While the curve of Fig. II-1 continues to rise very slowly until $R' = 115$, it must be remembered that it is based on a 65 F average temperature and that extension to layer thicknesses including the colder upper atmosphere would require a downward revision of the average temperature with a consequent lowering of the curve. In any case, the value of $R' = 105$ is not likely to be exceeded by very much when the basic atmospheric temperature at the surface is 70 F.

The black body radiation from a square foot of surface at 70 F is, by the Stefan-Bolzman Law,

$$R_b = 0.174 \times 10^{-8} T_1^4 = (0.174 \times 10^{-8})(530)^4 = 137 \text{ Btu per hour-ft}^2 \quad \text{.....(II-1)}$$

where T_1 is in degrees Fahrenheit absolute.

Hence the basic net outward radiation is

$$R_b - R_b' = \triangle R_b = 137 - 105 = 32 \text{ Btu per hour-ft}^2 \quad \text{.....(II-2)}$$

If the average temperature of origin in the first 5000 ft were 70 F instead of 65 F, R' would be approximately 110 Btu per hour-ft² as determined by an extrapolation of Hottel's curves. Hence the layer's emissivity would be 110/137 or 0.8. Based on this,

$$F' = 0.8F \quad \text{.....(II-3)}$$

Appendix III
Harmonic Analysis of Solar Intensity and Outdoor Air Temperature Curves

Any quantity that is a function of time and which varies in magnitude periodically in such a manner that each cycle of variation is identical with each other cycle can be expressed as a constant plus a series of sinusoidal variations of different frequencies. Such a series is called a Fourier series. Thus, if the function to be represented is

$$y = f(\theta) \quad \text{.....(III-1)}$$

the Fourier series is

$$y = \bar{y} + M_o Cos\,\omega_o\theta + N_o sin\,\omega_o\theta + M_1 Cos\,\omega_1\theta + N_1 sin\,\omega_1\theta + M_2 Cos\,\omega_2\theta + N_2 sin\,\omega_2\theta + \ldots \quad \text{.....(III-2)}$$

In (III-2) ω_o, ω_1, ω_2, etc., are the angular velocities corresponding to frequencies of the fundamental and the various harmonics of the sinusoidal variations. Expressed in radians,

$$\omega_o = \frac{2\pi}{\text{(time to complete one cycle of the fundamental)}}$$

and in degrees

$$\omega_o = \frac{360}{\text{(time to complete one cycle of the fundamental)}}$$

The other angular velocities are related to the fundamental as follows:

$$\omega_1 = 2\omega_o$$
$$\omega_2 = 3\omega_o$$
$$\omega_4 = 4\omega_o$$
etc.

The harmonic series can also be written as follows:

$$y = \bar{y} + \sqrt{M_o^2 + N_o^2}\cos(\omega_o\theta - \delta_o) + \sqrt{M_1^2 + N_1^2}\cos(\omega_1\theta - \delta_1) + \ldots \quad \text{(III-3)}$$

where

$$\tan\delta = \frac{N}{M} \quad \text{(III-4)}$$

and

sin δ has the sign of N

cos δ has the sign of M

Equation (III-3) is the form used in the main body of the paper to represent the variation in outdoor temperature and solar intensity.

In representing the outdoor air temperature and intensity of solar radiation on a surface it is assumed that these quantities have the same magnitudes at the same time of day on successive days. This is, of course, an approximation, but is sufficiently accurate for practical purposes. Furthermore, in any practical case it is rarely necessary to consider harmonics higher than the first, and often the fundamental plus the constant term gives a sufficient degree of accuracy, as pointed out in the main body of the paper.

The harmonic series representing a given variation in outdoor air temperature or solar intensity can be determined easily by carrying out the following steps. This procedure is illustrated by an example deriving the harmonic series for the solar intensity curve at Pittsburgh, Pa., on a horizontal surface, September 10, 1931,[3] as shown by Fig. 7a. Table III-1 gives the numerical example, which is referred to in the following step-by-step explanation.

Given a curve $y = f(\theta)$ over a period of one day:

1. Enter in the second column of the form shown in Table III-1 the hourly ordinates y of the curve to be analyzed.
2. In the third column, list the values of $\omega_o\theta$ for each hour, and in the fourth and sixth columns list the corresponding values of *cos* $\omega_o\theta$ and *sin* $\omega_o\theta$, respectively.
3. The fifth and seventh columns are filled in by multiplying each hourly ordinate y by the corresponding value of *cos* $\omega_o\theta$ and *sin* $\omega_o\theta$, respectively.
4. In the eighth column, list the values of $\omega_1\theta$ (which are twice as large as $\omega_o\theta$ at any given hour), and in the ninth and eleventh columns, respectively, list the corresponding values of *cos* $\omega_1\theta$ and *sin* $\omega_1\theta$.
5. The tenth and twelfth columns are filled in by multiplying each hourly ordinate y by the corresponding values of *cos* $\omega_1\theta$ and *sin* $\omega_1\theta$, respectively.
6. Add the ordinates in the second column and divide by 24; this gives the average ordinate of the curve, $\bar{y}$.
7. Add the values of y *cos* $\omega_o\theta$ in the fifth column and divide the sum by 12; this yields the coefficient M_o of the *cos* $\omega_o\theta$ term in the final expression for the curve.
8. Add the values of y *sin* $\omega_o\theta$ in the seventh column and divide the sum by 12; this yields the coefficient N_o of the *sin* $\omega_o\theta$ term in the final expression.
9. By repeating this procedure with the first harmonic tabulations in columns ten and twelve, coefficients M_1 and N_1 are obtained.
10. This method may be extended to any number of harmonics by setting up additional columns for $\omega\theta$, *cos* $\omega\theta$, *sin* $\omega\theta$, y *cos* $\omega\theta$ and y *sin* $\omega\theta$ in each harmonic desired.

The final expression for y is then given by (III-2). Whenever so desired, this can be reduced to the form of equation (III-3), determining the displacement angle δ from (III-4). This procedure is illustrated in Table III-1.

Table III-1—Harmonic Analysis of Intensity of Solar Radiation on Horizontal Surface at Pittsburgh, Pa., September 10, 1931

1	2	3	4	5	6	7	8	9	10	11	12
Time	y	$\omega_o t$, Deg	*cos* $\omega_o t$	y *cos* $\omega_o t$	*sin* $\omega_o t$	y *sin* $\omega_o t$	$\omega_1 t$, Deg	*cos* $\omega_1 t$	y *cos* $\omega_1 t$	*sin* $\omega_1 t$	y *sin* $\omega_1 t$
12 M	0	0	1.000	0	0	0	0	1.000	0	0	0
1 A.M.	0	15	0.966	0	0.259	0	30	0.866	0	0.500	0
2 A.M.	0	30	0.866	0	0.500	0	60	0.500	0	0.866	0
3 A.M.	0	45	0.707	0	0.707	0	90	0	0	1.000	0
4 A.M.	0	60	0.500	0	0.866	0	120	−0.500	0	0.866	0
5 A.M.	0	75	0.259	0	0.966	0	150	−0.866	0	0.500	0
6 A.M.	10	90	0	0	1.000	10	180	−1.000	−10	0	0
7 A.M.	20	105	−0.259	−5	0.966	19	210	−0.866	−17	−0.500	−10
8 A.M.	50	120	−0.500	−25	0.866	43	240	−0.500	−25	−0.866	−43
9 A.M.	105	135	−0.707	−74	0.707	74	270	0	0	−1.000	−105
10 A.M.	110	150	−0.866	−95	0.500	55	300	0.500	55	−0.866	−95
11 A.M.	110	165	−0.966	−106	0.259	28	330	0.866	95	−0.500	−55
12 N	110	180	−1.000	−110	0	0	360	1.000	110	0	0
1 P.M.	125	195	−0.966	−121	−0.259	32	30	0.866	108	0.500	63
2 P.M.	110	210	−0.866	−95	−0.500	−55	60	0.500	55	0.866	95
3 P.M.	75	225	−0.707	−53	−0.707	−53	90	0	0	1.000	75
4 P.M.	50	240	−0.500	−25	−0.866	−43	120	0.500	−25	0.866	43
5 P.M.	20	255	−0.259	−5	−0.966	−19	150	−0.866	−17	0.500	10
6 P.M.	0	270	0	0	−1.000	0	180	−1.000	0	0	0
7 P.M.	0	285	0.259	0	−0.966	0	210	−0.866	0	−0.500	0
8 P.M.	0	300	0.500	0	−0.866	0	240	−0.500	0	−0.866	0
9 P.M.	0	315	0.707	0	−0.707	0	270	0	0	−1.000	0
10 P.M.	0	330	0.866	0	−0.500	0	300	0.500	0	−0.866	0
11 P.M.	0	345	0.966	0	−0.259	0	330	0.866	0	−0.500	0
	895.0			−714.0		27.0			329.0		−22.0
$\bar{y} = 37.0$			$M_o = -59.6$		$N_o = 2.3$			$M_1 = 27.4$		$N_1 = -1.9$	

$\sqrt{M_o^2 + N_o^2} = 59.6$ | $\delta_o = \tan^{-1}\frac{N_o}{M_o} = 178$ | $\sqrt{M_1^2 + N_1^2} = 27.4$ | $\delta_1 = \tan^{-1}\frac{N_1}{M_1} = 356$

Thus, $y = 37 - 59.6\cos(15\Theta - 178) + 27.4\cos(30\Theta - 356) + \ldots$ where y is the intensity of solar radiation (I). This expression is equation (20) in the main body of the paper.

Incomes of Professions

A preliminary analysis of some of the results of a study of income from professional practice conducted over a period dating from 1929 has been published by the National Bureau of Economic Research. The report is an intensive analysis of data collected by the Department of Commerce in connection with its studies of national income.

The five professional groups covered are: physicians and surgeons, dentists, certified public accountants, lawyers and consulting engineers. The first section of the bulletin presents the average levels of net income in the five professions for a series of years. The most striking fact in this analysis is that the average income of consulting engineers, approximately a third higher than that of the next highest in the year 1929, drops sharply in three years to the lowest average income.

The average incomes of all the other four professions, while falling sharply with the exception of the dental profession, do not reach as low an average. In the report, this drop is noted as caused by "violent cyclical fluctuations in activity of the construction and heavy industries in general. And consulting engineers are in an even more vulnerable cyclical position than engineers as a whole, since their services are required in larger part in connection with the initiation of new projects or the expansion of existing enterprises."

CORRELATING THERMAL RESEARCH

As a part of the efforts of the ASHVE Research Laboratory to correlate research in thermal engineering carried on by the many institutions engaged in such work, and to disseminate the published results of such studies together with other reports of progress in the field, and in order to make this information available to the membership of the Society, there is published monthly on this page a limited number of brief abstracts of articles which it is believed will be of interest to all concerned. For more complete lists address the Librarian, ASHVE Research Laboratory, U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.

F. C. Houghten, *Director*.

W. L. Fleisher, *Chairman*
COMMITTEE ON RESEARCH

How to Overcome Condensation in Building Walls and Attics, by L. V. Teesdale. *Heating and Ventilating*, Vol. 36, No. 4, April 1939, pp. 36-40. Abstracts from paper presented before the Air Conditioning Conference at the University of Illinois, March 8-9, 1939. Recommendations for new construction and for remodeling existing homes to decrease condensation.

•

Design Calculations for Air Conditioning. *Heating and Ventilating*, Vol. 36, No. 3, March 1939, pp. 48-49 and 100. A typical problem is solved to illustrate the use of tables and charts in designing ducts for modern air conditioning systems.

•

Steam Heating System for High-Grade Homes, by T. N. Thomson. *Plumbing and Heating Journal*, Vol. 106, No. 4, April 1939, pp. 22-24 and 71-72. Description and illustrations of full layout particulars of a two-pipe steam plant including air conditioner and individual control at radiator valves, where desired, along with pressure control at the boiler.

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Comforto Termico, by Paulo Sa. *Instituto Nacional De Technologia*, Rio De Janeiro. The author studies different ways of measuring thermic comfort; dry- and wet-bulb thermometers, resultant thermometer of Missenard, effective temperature, Kata thermometer of Hill, globe thermometer, thermo-integrator, British equivalent temperature, relative humidity and velocity of the air. 60 pages.

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Ratings and Specifications on Attic Fans and Basement Blowers. *Air Conditioning and Oil Heat*, Vol. XII, No. 4, April 1939, p. 15. Table of ratings and specifications of attic fans and basement blowers produced by various manufacturers; also approximate retail price.

•

The Use of Thermocouples for Temperature Measurement, by Margaret Fishenden. *The Journal of the Institution of Heating and Ventilating Engineers*, Vol. 7, No. 73, March 1939, pp. 15-35. Discussion of use of thermocouples to measure temperatures and some of the precautions which should be taken to obtain reliable results. Effect of different alloys used in thermocouples. Care should be taken to see that the thermocouple junction is same temperature as gas to be measured. Also radiation between couple and walls should be noted.

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A Downdraft Conversion Burner for Domestic Furnaces, by Julian R. Fellows. *Mechanical Engineering*, Vol. 61, No. 4, April 1939, pp. 278-280. Description of downdraft conversion burner for use in a domestic furnace. Hand firing with or without automatic draft control will produce uniform temperature with smokeless conditions. Any coal except large lumps may be used but regular stoker coal is more satisfactory.

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Air Curtains, A Method for Controlling Open Door Drafts, by H. F. Purcell. *American Artisan*, Vol. 108, No. 4, April 1939, pp. 78-80. An air curtain or a high velocity air stream flowing in a given direction will form an invisible curtain which cannot be easily penetrated by an air stream with a lower velocity. Use of air curtain in heating an open door space.

•

Utilizing Heat from the Sun, by Dr. C. G. Abbot. *Smithsonian Miscellaneous Collections*, Vol. 98, No. 5. Description of some solar radiation boilers, water distillers and cookers. Methods of storing heat during cloudy weather and at night. Efficiency of solar flash boiler about 15 per cent. New Mexico could supply from solar radiation as much power yearly as is used from coal, oil and water at present.

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Calculation of the Summer Cooling Load. Forced Air Heating Facts from the Research Residence, by S. Konzo. *American Artisan*, Vol. 108, No. 4, April 1939, pp. 61-64 and 110-112. Detailed mathematical discussion of calculation of summer cooling load for a residence or larger building. Indoor design temperatures and humidities. Cooling load divided into 11 items—heat transmission through walls, windows, floors; heat given off by occupants, etc.

•

Summer Cooling in the Warm-Air Heating Research Residence with Cold Water, by A. P. Kratz, S. Konzo, M. K. Fahnestock, and E. L. Broderick. *University of Illinois Bulletin*, Vol. XXXV, No. 101, August 16, 1938, Bulletin No. 305.

•

Some Preliminary Observations on the Effects of Sustained Muscular Effort in a Hot Climate with Special Reference to the Loss of Fluids and Chlorides, by N. C. Lendon. *Journal Royal Army Medical Corps* (London), Vol. 71, Nov. 1938, pp. 318-324. Result of observations made at Peshawar, India. The subjects were three soldiers with less than six months and three soldiers with more than five years service in the tropics. Loss of fluids and chlorine and other physiological reactions during nine-mile march, with shade temperature of 104 F and relative humidity of 47 per cent; during 3 hours, 10 min, 6 pints fluid loss and 6.6 gr Cl.

•

Bioclimatics:—A Science of Life and Climate Relations, by Andrew Delmar Hopkins. *U. S. Department of Agriculture, Miscellaneous Publications, No. 280 M*, 1938, 166 pages. A summary of results of over 40 years of personal observation of life and climate and conclusions drawn therefrom. Derivation of certain basic principles and application of the solution of particular problems. Bioclimatic law for comparison of data. By this law phenomena of life and climate should be equal under equal influences along lines called isoplanes.

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Thermal Insulation of Glass and Plate Glass, by Joseph Carlier. *Verre and Silicates Ind.*, Vol. 9, No. 12, pp. 136-137, 1938. To obtain a lower coefficient of thermic transmission of glass the author recommends covering one side of the glass with a plastic insulating mix, transparent or translucid and protected from the humidity of the air, of a solution of phenolformolic azoline. This resin does not harden when heated. A thickness of 1.5 mm of resin between two panes of glass lowers the coefficient of thermic transmission to 2.6.

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Glass in Construction and Decoration. A Review of Recent Developments, by Ramond McGrath. *The Architectural Review Supplement*, February 1939, pp. 99-108. Discussion of methods of using glass as decorative materials in buildings such as colored glass and glass blocks. Illustrations of glass blocks used with masonry construction. Refers to data supplied by Building Research Station, England, entitled Insulight Glass Masonry. These reports cover light transmission, and distribution, heat transmission, strength and fire resistance of glass blocks. Many illustrations.

Tunnel Ventilation Described to Massachusetts Chapter

May 23, 1939. The May meeting of Massachusetts Chapter concluded another successful year. Chapter members were guests of the Harvard School of Public Health and the speaker of the evening was A. L. O'Banion, engineer, Sumner Tunnel, City of Boston.

Mr. O'Banion discussed the Ventilation and Operation of the Sumner Tunnel, of which the construction and ventilation were amply illustrated by slides. Many questions were asked at the close, which were answered in an interesting manner by the speaker.

Refreshments were served after the business meeting.

Prior to the regular Chapter meeting the Board of Governors elected the following officers for the coming year:

President—H. C. Moore.
Vice-President—C. P. Yaglou.
Secretary-Treasurer—C. M. F. Peterson.
Councillors to Engineering Societies of New England: H. C. Moore, Alfred Kellogg.

April 25, 1939. A meeting of Massachusetts Chapter was held at Massachusetts Institute of Technology and was in charge of R. M. Nee, vice-president of the Chapter.

The subject of the meeting was Steam Utilization: Engineering and Control and a discussion of this timely subject was carried on by Mr. Nee of the Boston Edison Co. and his assistants. The comments showed how much valuable work has been done and that much more can be done concerning the economical use of steam.

Nominations for the Board of Governors for the coming year were announced as follows: D. J. Edwards, C. M. F. Peterson, C. P. Yaglou.

March 21, 1939. Massachusetts Chapter members met at Massachusetts Institute of Technology with more than 50 members and friends present.

J. H. Van Alsburg, Chicago, Ill., a member of the ASHVE Committee on Research, was guest speaker and addressed the meeting on the subject of the Four D's of Air Conditioning—Ducts, Dampers, Distribution and Deflection. Mr. Van Alsburg not only covered the subjects of his talk, but also gave an interesting account of the projects being carried out in the ASHVE Research Laboratory and cooperating institutions. As a result of Mr. Van Alsburg's talk, Secy. H. C. Moore reports that every member went away with a better understanding of the importance of the research work being undertaken by the Society.

Southern California Chapter Officers

May 16, 1939. Southern California Chapter held a dinner meeting at the Royal Palms Hotel, Los Angeles.

Pres. H. M. Hendrickson conducted a short business meeting, after which the following officers were elected to serve for the coming year:

President—J. F. Park.
Vice-President—H. H. Douglas.
Secretary—E. P. Wells.
Treasurer—Glen Miller.
Board of Governors—H. M. Hendrickson, A. J. Hess, W. D. Fabling, O. W. Ott.

Following the elections, a film was presented, entitled Underwriters' Laboratory Approved.

The speaker of the evening was Prof. B. F. Raber, University of California, Berkeley, who spoke on the subject of Evaluating Air Conditioning Bids. He presented a method of evaluating bids from bidders on jobs for which there were no plans or specifications prepared in advance. The talk was followed by a lively discussion which indicated the interest of the members in this subject, according to the report of A. J. Hess.

April 11, 1939. The meeting of Southern California Chapter was held at the Colonial Banquet Room at 7:45 p. m. after an informal dinner. The occasion of the meeting was the visit of J. F. McIntire, Detroit, president of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Following the completion of routine business, President McIntire addressed the Chapter on the subject of The Birth of Air Conditioning and carried on an informal discussion concerning the affairs of the Society.

Secy. A. J. Hess has reported that the Southern California Chapter very much enjoyed the visit of President McIntire and it is their sincere hope that some day in the near future he will be able to return and visit them again.

Philadelphia Chapter Enjoys Golf Outing

May 10, 1939. The regular meeting and annual golf outing of the Philadelphia Chapter was held at the St. Davids Country Club, Wayne, Pa. The Chapter was fortunate in having good weather and 63 members and guests played golf.

A very enjoyable dinner was served at the Club, which was followed by a short business meeting. New officers for the ensuing year, who were installed, were:

President—R. F. Hunger.
First Vice-President—C. B. Eastman.
Second Vice-President—H. B. Hedges.
Secretary—Edwin Elliot.
Treasurer—H. H. Mather.
Board of Governors—R. F. Hunger, H. H. Erickson, C. B. Eastman, H. B. Hedges, H. H. Mather, Edwin Elliot, M. G. Kershaw.

A number of awards were made for outstanding performance on the golf course during dinner. In the Kicker's Handicap, the following Chapter members were among the winners: C. W. Coward 81, F. J. Wilson 80, C. B. Eastman 84; low gross, W. P. Culbert 79, A. P. Shanklin 86. The Chapter guests who received awards were: Joseph Stecher 84; G. P. Alexander 84; low gross, E. C. Powers 71; Leo Garrity 86. Mr. Culbert was chairman of the Golf Committee.

R. F. Hunger presented a smoking set to retiring president H. H. Erickson on behalf of the Chapter as a token of appreciation for his outstanding leadership during the past year.

The meeting adjourned at 9:30 p. m. according to the report of H. H. Mather.

April 13, 1939. The regular meeting of Philadelphia Chapter ASHVE was held at the Engineers Club. Minutes of the previous meeting and the treasurer's report were read and approved.

A verbal report was rendered by C. F. Dietz, chairman of the Auditing Committee, stating that this committee had been over the books of the Chapter and had found them to be in order.

Edwin Elliot, chairman of the Membership Committee, made a complete report in connection with Chapter membership.

Details of the Semi-Annual Meeting of the Society at Mackinac Island, July 4-6, were presented by Mr. Blankin.

President Erickson read the nominations made by the Nominating Committee for officers of the Chapter for the coming year. President Erickson called for further nominations from the floor, but as none were made a vote was taken to instruct Secy. H. H. Mather to cast a ballot for the election of officers.

The members present listened to an entertaining talk by W. A. Grant of the Carrier Corp. on Zoning of Air Conditioning Systems. Following a rising vote of thanks to Mr. Grant at the conclusion of his paper, the meeting was adjourned at 9:30 p. m.

Meetings of Washington, D. C., Chapter

May 10, 1939. The annual meeting of Washington, D. C. Chapter was held at the Dodge Hotel with an attendance of approximately 50 members and guests. Minutes of the April 10 and 12 meetings were read and approved.

W. E. Kingswell, M. F. Hoppe and Mr. Burns were appointed tellers.

While the ballots were being counted, the film Cold Magic was shown, which was made available through W. R. Lockhart and J. M. Lambert.

When the tellers reported, announcement was made of the election of the following officers:

President—T. H. Urdahl.
Vice-President—F. E. Spurney.
Secretary—P. H. Loughran, Jr.
Treasurer—G. R. Walz.
Board of Governors—S. P. Eagleton, S. L. Gregg, E. V. Fineran.

The members then visited the Capitol power plant and the House office and Capitol air conditioning systems through the courtesy of G. D. Fife.

April 12, 1939. The seventh regular monthly meeting of the season was held in the North Interior Building, Washington, D. C. The attendance was approximately 30 members and guests.

S. L. Gregg made an announcement of the Semi-Annual Meeting of the Society at Mackinac Island, July 4-6.

The report of the Nominating Committee offered the slate of officers to be elected at the May 10 meeting and no additional nominations were received from the floor.

J. M. Lambert, refrigeration engineer, gave an illustrated talk on Cold Storage Lockers. The great interest that this talk aroused was evidenced by the long discussion which followed, according to the report of Secy. E. V. Fineran.

April 10, 1939. At the invitation of the *Baltimore Air Conditioning Society* the Washington, D. C. Chapter met at the Engineers Club in Baltimore. The members of the Baltimore Society proved to be most gracious hosts. There were refreshments and then a splendid dinner, following which a unique system was effectively employed to introduce the members and guests present.

According to Secy. E. V. Fineran after-dinner jokes were above par. Even during the intermission ASHVE members were entertained at ping pong, although one member was on the wrong end of a 21 to 3 score.

The meeting was then addressed by Prof. W. F. Wells, University of Pennsylvania, on the subject of Sanitary Ventilation with Emphasis on Removal of Micro-organisms from Indoor Air.

March 8, 1939. Approximately 50 members and guests attended a meeting of Washington, D. C. Chapter at the Dodge Hotel. Minutes of the February meeting were read and approved.

Pres. S. P. Eagleton announced the appointment of F. A. Leser as chairman of the Membership Committee to succeed E. H. Lloyd, who resigned.

The following members were appointed to the Nominating Committee: F. M. Thuney, W. H. Littleford, P. H. Loughran, Jr.

A request for members to bring their dues up-to-date was made by W. R. Lockhart.

F. E. Spurney, chairman of the Program Committee, announced that there would be a joint meeting with the *Baltimore Air Conditioning Society*. It was suggested that members arrange among themselves for group transportation.

The Chapter was advised that Prof. A. G. Christie, president of the *ASME*, was to present an illustrated lecture on Modern Steam Generators at Catholic University on March 9.

T. H. Urdahl, chairman of the Steering Committee, then presented for general discussion the suggested Ordinance Governing the Installation of Air Conditioning, Warm Air Heating, Air Cooling and Ventilating Systems Employing Ducts.

New York Chapter Social Meeting

May 15, 1939. The New York Chapter closed the 1938-39 season with its annual stag social meeting at the Building Trades Club, New York City.

Following dinner, the speaker of the evening, Bob Willis, humorist of WOR broadcasting station, was introduced and brought into play all of his better stories. Repertoires of the members were greatly enriched by the ones they were able to remember from rapid bombardment. After Mr. Willis concluded, the meeting broke up into smaller groups, who proceeded to enjoy the remainder of the evening playing cards, singing with an accordian player, and playing pool.

Officers for the year 1939-40 were announced as follows:

President—O. O. Oakes.
Vice-President—C. S. Pabst.
Secretary—T. W. Reynolds.
Treasurer—W. M. Heebner.
Board of Governors—H. G. Meinke, A. L. Browne, H. L. Baker, Jr.

According to the report of Mr. Baker, the entire chapter is looking forward to an active and progressive year under the leadership of these new officers.

Pres. McIntire Addresses Manitoba-Minnesota Members

May 13, 1939. The annual meeting of Manitoba Chapter was held at the Edgewater Beach Hotel, Detroit Lakes, Minn., on the occasion of a joint meeting of the Minnesota and Manitoba Chapters.

Pres. William Worton presided during the business session with the following members present: E. J. Argue, J. B. Steele, G. C. Davis, William Glass, P. L. Charles, Ivan McDonald, and two guests—F. Chester and O. J. Hatch.

After an enjoyable dinner with wives and members present, there was a short business meeting and the following officers for 1939-40 were nominated for the Manitoba Chapter and elected:

President—G. C. Davis.
Vice-President—P. L. Charles.
Secretary—Ivan McDonald.
Treasurer—H. R. Eade.
Board of Governors—William Glass, Frank Thompson, William Worton.

Mr. Worton called on the new president, who gave a short address.

It was then moved by Mr. Worton that further business be adjourned until the next meeting and this was seconded by Mr. Charles.

J. F. McIntire, Detroit, president of the ASHVE, was guest of honor and gave a short address to the entire group relative to his contacts with the various chapters of the Society.

The meeting was then adjourned for a social evening, according to the report of Mr. Davis.

Michigan Chapter Officers Elected

May 23, 1939. The annual golf meeting and election of officers of Michigan Chapter was held at the Detroit Golf Club, where there were 36 who played golf and 45 who attended the dinner and evening meeting.

The meeting was called to order at 9:00 p m. by Pres. F. J. Linsenmeyer. Roll call and the reports of the secretary and treasurer were called for. President Linsenmeyer gave a short resumé of meetings during the year and thanked the officers and committee members for the cooperation given him during his administration. He appointed F. X. Marzolf and Matthew Soeters as an Auditing Committee to audit the treasurer's report.

The Semi-Annual Meeting of the Society at Mackinac Island, Mich., July 4-6, was discussed by W. G. Boales, general chairman of the Committee on Arrangements.

President Linsenmeyer then called on the chairman of the Nominating Committee, F. J. Feely, who presented the following list of nominations for 1939-40:

President—W. C. Randall.
Vice-President—G. H. Tuttle.
Secretary—W. H. Old.
Treasurer—J. S. Kilner.
Board of Governors—F. R. Bishop, R. F. Connell, M. C. Kincaide, J. S. O'Gorman, Jr., S. S. Sanford, M. B. Shea.

There were no further nominations from the floor, and it was moved by G. D. Winans and seconded by E. W. Gifford that the nominations be closed. This motion was passed.

It was moved by F. R. Bishop, and seconded by Mr. Gifford that the secretary cast a unanimous ballot for Mr. Randall for president, Mr. Tuttle for vice-president, Mr. Old for secretary, and Mr. Kilner for treasurer. This motion was passed.

Robert Clar, Jr. and J. T. Reader were appointed as tellers for the election of members to the Board of Governors. The ballot-

ing resulted in the election of Messrs. Shea, Sanford and O'Gorman.

Mr. Winans, chairman of the Golf Committee, announced the following winners of the golf tournament: high gross—Mr. Emerick, low gross—Mr. Wood.

In the Kickers' Handicap, awards were made to Messrs. E. W. Gifford, W. G. Boales, E. M. Harrigan, W. H. Old, Robert Clar, Jr., R. K. Milward and E. D. Sheley.

Professor Linsenmeyer then turned the meeting over to the newly-elected president, Mr. Randall, who thanked the membership for the honor bestowed upon him and introduced the new officers and members of the Board of Governors.

The meeting adjourned at 11:30 p. m.

April 17, 1939. Michigan Chapter, met at The L'Aiglon, in the Fisher Building, with approximately 60 members and guests present.

C. T. Olmsted, secretary of the State Board of Registration of Professional Engineers, discussed the recently enacted Registration Act. He pointed out that in order to register without examination, an applicant must have had 12 years experience of a responsible character, exclusive of college, prior to January 1, 1938.

The second guest speaker was F. R. Bichowsky, consulting engineer, Dow Chemical Co., Ann Arbor, Mich. Mr. Bichowsky discussed the subject of Dehumidification in Air Conditioning with particular reference to chemical dehumidification for industrial applications. He described some of the advantages and disadvantages of silica gel, lithium chloride, activated alumina and calcium chloride.

H. F. Hutzel, Kelvinator Corp., C. F. Cushing, Bryant Heater Co., and R. B. Derr, Aluminum Co. of America, discussed Mr. Bichowsky's paper, according to the report of Secy. G. H. Tuttle.

Illinois Chapter Discusses Chicago Building Code and Air Conditioning Markets

April 10, 1939. A meeting of Illinois Chapter was called to order by Pres. J. R. Vernon with 124 members and guests present, which was the largest attendance of the year.

After the minutes of the March meeting were dispensed with, I. E. Brooke, chairman of the Membership Committee, introduced two new members, W. R. Blayney and A. S. Ammerman, Jr.

President Vernon reminded the gathering of the Society's Semi-Annual Meeting at Mackinac Island, Mich., July 4-6, and Treas. E. M. Mittendorff announced that a few members owed dues for 1938-39.

J. H. Milliken moved that the question of Illinois Chapter membership in Illinois Engineering Council, which was tabled at the March meeting, be again referred to the Board of Governors for study and recommendation to the Chapter. The motion was seconded and carried.

This meeting was announced as an open and informal discussion of the new Chicago Building Code with special reference to those parts of the code pertaining to heating, ventilation and air conditioning. John Howatt, past president of the ASHVE, presided and introduced the various authorities who were to take part in the discussion.

The Chapter had invited the members of the *Illinois Society of Architects* and Elmer Jensen, president of this group, opened the meeting with a brief resumé of the forming of the new code. G. C. Gearon represented the Department of Boiler Inspection of the City of Chicago. J. J. Aeberly represented the Chicago Board of Health, John Neale the Underwriter's Laboratory, and J. T. W. Babcock the Western Factory Insurance.

Slides were used to illustrate pertinent portions of the code, and as each slide was shown, it was discussed briefly by Chairman Howatt and any questions from the audience were answered by one of the speakers. The fact that there is a great deal of interest in this subject was evidenced by the large number of questions and the detailed discussions which followed the showing of slides illustrating portions of the code.

Among those taking part in these discussions were Messrs. I. E. Brooke, T. F. Hanley, Jr., H. M. Hart, R. E. Hattis, Tom Brown, H. G. Chapin, S. R. Lewis, J. C. Hines and M. W. Bishop. Many interesting points were discussed and everyone felt that the meeting had been well worthwhile from every viewpoint.

The meeting closed on a sad note, as word was received of the death of B. F. Baker, chairman of the board of Kewanee Boiler Corp. It was moved and seconded that a resolution expressing the Chapter's sorrow and sympathy be spread on the Chapter records and that a copy be sent to Mrs. Baker.

The meeting was adjourned at 10:00 p. m., according to the report of Secy. M. W. Bishop.

March 13, 1939. The March meeting of Illinois Chapter was called to order at 7:45 p. m. by Pres. J. R. Vernon with 92 members and guests attending. The minutes of the February meeting were read by Secy. M. W. Bishop and approved as read.

President Vernon appointed Messrs. G. G. Turner, W. W. Voss and C. A. Gustafson as tellers to count the ballots for the Nominating Committee.

Tom Brown spoke of the Semi-Annual Meeting of the Society at Mackinac Island, Mich., July 4-6, and announced the appointment of A. O. May as chairman of attendance of Illinois Chapter for this meeting.

Membership Chairman I. E. Brooke introduced two new members, J. S. Kearney and M. L. Buckley, and Meetings Chairman V. L. Sherman announced the details of the April meeting.

President Vernon called on C. M. Burnam, Jr., who introduced the speakers of the evening, K. C. Porter and William Rock, both of the Commonwealth Edison Co. Their subject was Air Conditioning Markets and Their Trends.

Mr. Porter's remarks dealt with an analysis of air conditioning markets and loads during the past six years in Chicago. Basing his remarks upon complete records of the utility company, the installations were divided into two general groups, Small Commercial, and Industrial, Office and Public Assembly.

The Small Commercial classification included restaurants as the largest single market and also included five and ten cent stores, shoe and drug stores, funeral parlors, and miscellaneous others. This last included many types which as a group show an appreciable tonnage but individually are not of sufficient size to consider. The other group Industrial, etc. included large department stores, offices, theaters, and industrial applications.

The speaker showed curves to indicate the trends of these classifications since 1932. For example, the Small Commercial classification showed that 26 per cent less tonnage was installed during 1938 than during 1937, while Industrial, etc. showed 61 per cent less in 1938 than in 1937.

Further breakdown of the two general groups showed that the four largest single types of installation in order of importance in 1938 were restaurant, general office, theater, and industrial. In 1932 industrial was the largest and restaurant was the smallest of the four.

Finally, Mr. Porter pointed out that the six-year total trend of all air conditioning markets had a fairly close relation to the curve of general business conditions for the same period. Over this time the Small Commercial curve was much more stable than the total air conditioning business curve.

Mr. Rock then took over the meeting and his talk included a forecast of future trends of air conditioning markets based on past records and apparent saturation of the market. Slides were shown which indicated the average hours of equipment usage per summer period for different types of establishments. A few of these were: beauty parlors, 553 hours; funeral parlors, 310 hours; general offices, 667 hours; restaurants, 942 hours. Drug stores showed the highest with 1330 hours and funeral parlors the lowest.

One slide of particular interest had 2500 air conditioned establishments indicated on a map of Chicago. Study of this map would indicate areas which were most likely fields for further

installations and which offer best possibilities for results from sales efforts. Another slide showed the per cent of saturation in the various markets based on the probable number of eventual installations rather than the number of establishments. A few of these figures were: theaters, 76.0 per cent; restaurants, 16.3 per cent; and beauty parlors, 1.3 per cent.

A general discussion of the points brought out by the speakers followed in which Messrs. I. E. Brooke, A. G. Sutcliffe, F. I. Raymond, R. E. Hattis, H. G. Chapin, M. W. Bishop and others took part.

President Vernon, on behalf of the Chapter, expressed appreciation of the excellent data made available by the speakers and the members gave them a rising vote of thanks.

G. G. Turner reported the result of the ballot for members of the Nominating Committee as follows: J. J. Aeberly, John Howatt, J. R. Vernon, J. J. Hayes, R. E. Hattis, Tom Brown and M. W. Bishop.

Development of Aluminum Reviewed at Pittsburgh Meeting

May 8, 1939. The meeting of Pittsburgh Chapter was called to order at 7:30 p. m. by Vice-Pres. F. C. McIntosh in the private dining room of Stouffer's Restaurant. Minutes of the April meeting were read and approved. Thirty-five members and guests were present.

R. B. Stanger, chairman of the Entertainment Committee, reported on arrangements for the June party.

The following committees presented final reports, after which the Chair directed that all committees be discharged: J. F. S. Collins, Jr., Annual Meeting Committee; E. C. Smyers, Auditing Committee; T. F. Rockwell, *NBFU* Code Committee.

When new business was taken up, a resolution was proposed by Mr. Collins as follows and was adopted:

Be It Resolved, That the Pittsburgh Chapter of the ASHVE extend its congratulations and best wishes to the newly formed Carnegie Tech Student Chapter of the Society, and extend a standing invitation to the members of the Carnegie Tech Student Chapter to attend all technical sessions of the Pittsburgh Chapter.

It was moved by Mr. Collins and passed that Secy. T. F. Rockwell be directed to send a letter of thanks to R. A. Miller and the Pittsburgh Plate Glass Co. for the two prizes given for the Annual Meeting of the Society.

G. G. Waters introduced P. V. Faragher, metallurgist, Aluminum Co. of America, as the evening's speaker. Mr. Faragher's subject was Scientific Development of Aluminum and in his introduction he described how abundant aluminum compounds were but how difficult it was to reduce them to the metallic state. He also reviewed the early experiments of Hall and others which finally led to the electrolytic method now used. The speaker then continued by discussing the properties of the common aluminum alloys. The program was concluded by showing a film which traced the production and manufacture of aluminum from the mine to a wide variety of finished products.

The meeting adjourned at 9:30 p. m.

Minnesota Members Enjoy Joint Meeting and Golf

June 12, 1939. A golf tournament at the Golden Valley Golf Club was held by Minnesota Chapter with approximately four foursomes taking part in the golf activities in the afternoon. Twenty-two members were present at dinner which was served at 7:30 p. m.

After dinner, Pres. J. E. Swenson called the meeting to order for election of officers and the following were elected for the coming year:

President—F. C. Winterer
Vice-President—M. H. Bjerken
Secretary-Treasurer—H. M. Betts
Board of Governors—J. E. Swenson and R. E. Backstrom

During the meeting golf awards were made to the winners with F. H. Schernbeck in charge of this part of the program.

No further business was transacted and the meeting was adjourned. A very pleasant evening was spent at cards, etc., according to the report of Mr. Bjerken.

May 13-14, 1939. Minnesota Chapter met jointly with the Manitoba Chapter at Edgewater Beach Hotel, Detroit Lakes, Minn. The honored guest was J. F. McIntire, Detroit, president of the ASHVE, who was accompanied by Mrs. McIntire. Twenty members, wives and guests from Winnipeg were present, with 30 members, wives and friends from Minnesota attending.

Saturday afternoon was devoted by the women to bridge, while the men availed themselves of the facilities for golf.

A banquet was served to 50 Saturday evening and, in the absence of Pres. J. E. Swenson and Vice-Pres. F. C. Winterer of Minnesota Chapter, N. D. Adams, member of the ASHVE Council, acted as chairman and extended greetings to the group. The meeting was then turned over to William Worton, president of Manitoba Chapter, who presided during the election of officers for Manitoba Chapter. A splendid address was given by President McIntire, as well as many interesting comments by various members of the Manitoba Chapter. An enjoyable dance followed the banquet.

G. C. Davis, who was elected president of Manitoba Chapter, presented the program chairman, M. H. Bjerken, Minneapolis, with a token consisting of the official badge used by all Canadians during the visit of King George and Queen Elizabeth. Mrs. Bjerken was presented with a beautiful program containing the activities during the visit of the King and Queen to the City of Winnipeg.

On Sunday morning, breakfast was served to 44 guests, and this was followed by a continuation of Saturday's golf game. There were also sightseeing tours around the lake district. Members departed for home in the late afternoon, after seeing President McIntire off for Chicago.

The meeting as a whole was most successful, according to Secretary Bjerken.

Annual Meeting of Ontario Chapter

May 1, 1939. The annual meeting of Ontario Chapter was held at the Royal York Hotel, Toronto, at 6:30 p. m. with 39 members present. Secy. H. R. Roth read the minutes of the preceding meeting, which were accepted as written.

C. Tasker announced the joint meeting of the *Affiliated Engineering and Allied Societies* in Ontario in the Concert Hall of the Royal York Hotel, Toronto, on May 18, when I. I. Sikorsky was the principal speaker.

J. H. Fox, chairman of the committee for the Society's Semi-Annual Meeting at Mackinac Island, reported on plans for transportation, entertainment, and the technical sessions.

The treasurer read the auditors' report on the financial standing of the Ontario Chapter as of April 29, with a balance on hand of $384.50. W. C. Kelly, one of the auditors, reported that 21 members were delinquent for two years. He moved that the Board of Governors make a concerted effort to obtain back dues from these members, and the motion was carried.

Mr. Kelly moved that the Ontario Chapter pay $15.00 in stenographer's fees for the past year and an additional $15.00 for fees not paid in 1938, and this motion was also carried.

Mr. Fox, chairman of the Attendance Committee, reported that during the past year the average attendance at meetings was 64 members and guests.

Mr. Tasker, chairman of the Papers Committee, reported on the programs during the past year and asked cooperation in arranging programs for the coming season from Chapter members.

W. R. Blackhall advised that the Ontario Chapter's University of Toronto award of $25.00 was given to A. E. Smith, who

wrote a thesis on Mechanical Draft. In addition a copy of THE 1939 GUIDE was given to F. D. Ledgett and C. R. Phillips as second and third awards.

In the absence of S. W. Alexander, chairman of the Membership Committee, Secretary Roth reported that during the past year 11 new members have joined the Chapter. As of April 29 the Chapter membership was 105.

M. W. Shears reported that the new revised Code on Steam and Hot Water Systems for Residences up to 50,000 cu ft capacity is now available, at 50 cents per copy to members and $1.00 per copy to non-members.

A. G. Ritchie, chairman of the Greeters Committee, reported on the past year's activities.

In the absence of E. R. Gauley, chairman of the Entertainment Committee, Pres. H. B. Jenney presented a statement on the committee's work and especially thanked the press for their splendid cooperation.

Secretary Roth re-read the report of the Nominating Committee for officers for the coming season. It was moved by Mr. Shears, seconded by J. W. Bishop, and carried that the secretary cast one vote, unanimously electing the following officers:

President—H. D. Henion
Vice-President—J. W. O'Neill
Secretary—H. R. Roth
Board of Governors—C. Tasker, F. E. Ellis, J. H. Fox.

Mr. Jenney as past president will automatically be a member of the Board of Governors for the coming year.

Mr. Jenney thanked the various committees for their splendid cooperation during the past year and turned the chair over to the newly elected president, Mr. Henion.

After making inquiry regarding the desirability of holding a golf meeting in June, President Henion announced that this would be arranged by popular demand.

St. Louis Chapter Elects Officers

May 2, 1939. Thirty members and guests of St. Louis Chapter met at the Kingsway Hotel for dinner. Pres. E. E. Carlson introduced and welcomed the visitors and then called on C. E. Hartwein, chairman of the Entertainment Committee, who stated that final arrangements had been made for the annual dinner dance to be held on June 23 at Norwood Country Club.

The treasurer of the Chapter reported a balance of $456.37.

The following committee chairmen were called on and submitted reports: W. J. Oonk, Publicity Committee; G. B. Rodenheiser, Educational Committee; and L. R. Szombathy, Membership Committee.

President Carlson requested Secy. D. J. Fagin to read the report of the Nominating Committee and stated that other nominations would be received from the floor. Paul Sodemann moved that the Secretary be instructed to cast a ballot for the Chapter accepting the report of the Nominating Committee and this was seconded by R. C. Driemeyer. The following officers were elected:

President—R. J. Tenkonohy
First Vice-President—C. E. Hartwein
Second Vice-President—D. J. Fagin
Secretary—C. F. Boester
Treasurer—M. F. Carlock
Board of Governors—E. E. Carlson, G. B. Rodenheiser, F. C. Laufketter and J. H. Carter.

The meeting was then turned over to Mr. Tenkonohy, chairman of the Program Committee, who introduced Clifford Greve, Jr., attorney at law, St. Louis, whose subject was Legal Phases of Selling and Contracting. Mr. Greve's talk proved to be very interesting and educational. He answered a number of questions regarding contracts, liabilities, etc.

Before adjourning the meeting Mr. Carlson thanked all those who had helped make his administration successful and stated that he stood ready to help the incoming officers in their respective duties. He then turned the gavel over to the newly elected president, Mr. Tenkonohy.

Death of R. E. Keyes

R. E. Keyes, chief engineer, B. F. Sturtevant Co., Cooling and Air Conditioning Division, Boston, Mass., died at the Metropolitan Hospital, Philadelphia, on May 29, 1939, of pneumonia.

Mr. Keyes was born September 19, 1885, at Keene, N. H., and was graduated in 1907 from Massachusetts Institute of

R. E. Keyes

Technology, Boston, with the degree of bachelor of science from the department of mechanical engineering. He was first employed by B. F. Sturtevant Co. in 1909 and was active in the development of air conditioning as applied to both human comfort and industrial air conditioning systems. During his business career he had also been associated with Cambridge Iron Works, Cambridge, Mass., as mechanical engineer, with Ilg Electric Ventilating Co. as eastern manager, with W. L. Fleisher & Co., Inc., New York, N. Y., as construction engineer, with Drying Systems, Inc., Chicago, and with Surface Combustion Corp., Toledo. Since 1926 he acted as chief engineer of The Cooling & Air Conditioning Corp., which became a division of the B. F. Sturtevant Co. in December, 1937.

Mr. Keyes made a fundamental contribution to the air conditioning industry in his basic development of the by-pass principles of by-passing recirculated air from the conditioned area around the conditioner and then mixing the two air streams, all under automatic control. He also developed and patented other system forms and controls and combinations of apparatus to effect close control of conditioned air, in all cases with a view to economy of operation.

In 1913 Mr. Keyes was elected a Member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. He assisted the Guide Publication Committee in the preparation of the HEATING, VENTILATING, AIR CONDITIONING GUIDE and at the time of his death he was a member of the Research Technical Advisory Committee on Sensations of Comfort. The Society has lost a loyal and active member.

Mr. Keyes is survived by his widow, Mrs. Maud E. Keyes of Wellesley Hills, Mass., to whom the Officers and Council extend their sincere sympathy

H. J. Ryan Dies

The death of H. J. Ryan, Albany, N. Y., on June 13, 1939, is recorded with sorrow for he had been a Member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS since 1922.

Mr. Ryan was born at Cohoes, N. Y., on December 29, 1880. He had a long training and experience in the heating and ventilating field. For 10 years he served as journeyman steamfitter and for 12 years he was a contracting steamfitter at Troy, N. Y. Following this he was engaged in sales activities with the Gurney Heater Mfg. Co., New York, N. Y. In 1922 he was a manufacturers' agent located at Albany, N. Y., representing the National Radiator Co., American District Steam Co., and the Graff Furnace Co. In 1923 Mr. Ryan was appointed representative in charge of the Albany office of The Trane Co., a position which he held at the time of his death.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 26 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by July 15, 1939, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
ATHERTON, G. R., Mgr., Vento Dept., American Radiator & Standard Sanitary Corp., Chicago, Ill. (*Reinstatement*)	John Howatt S. R. Lewis	Albert Buenger C. V. Haynes
AUER, G. G., Pres., The Auer Register Co., Cleveland, Ohio	L. T. Avery C. M. H. Kaercher	J. E. Wilhelm L. E. Slawson
BELL, S. R., Consulting Engr., Sydney R. Bell & Associates, Melbourne, Australia.	G. I. Davey T. A. Roseby	Gerald Haskins (*ICE*) R. V. O. Wells (*ICE*)
BERGER, J. L., Secy., W. R. Rhoton Co., Cleveland, Ohio	W. R. Rhoton D. L. Taze	W. M. Rowe L. T. Avery
BROWN, S. D., Office Mgr. & Sales Engr., The Trane Co., Spokane, Wash.	E. C. Willey J. P. McDermott	J. D. Kroeker M. C. Phillips (*Non-Member*)
CARROLL, E. E., Owner, Kleenair Furnace Co., Portland, Ore.	B. W. Farnes C. E. Heinkel	J. D. Kroeker T. E. Taylor
COLBY, J. H., Sales Engr., Johnson Service Co., Boston, Mass.	James Holt H. C. Moore	H. B. Wiegner M. F. Rather
COOPER, D. E., Partner, D. E. Cooper & Son, Salem, Ore.	E. C. Willey C. E. Heinkel	B. W. Farnes H. H. Stanton (*Non-Member*)
CULLEN, A. C., Executive, Cullen Co., Washington, D. C. (*Advancement*)	H. R. Iverson L. F. Nordine	T. H. Urdahl W. A. Danielson
FOLEY, D. F., Asst. Mgr., W. B. Young Supply Co., Kansas City, Mo. (*Advancement*)	F. F. Dodds H. H. Wright	M. M. Rivard W. A. Russell
GHOSH, B. B., Air Cond. Engr., Messrs. Refrigerators (India) Ltd., Calcutta, India.	T. A. Curry (*ICE*) R. Wolfenden (*IME*)	T. N. Banerjea (*RSI*) N. L. Hindley (*ICE*)
HALL, N. H., Engr., East Ohio Gas Co., Cleveland, Ohio	L. T. Avery W. E. Stark	C. M. H. Kaercher C. F. Cushing
HEIKKILA, F. E., Sales Engr., Westinghouse Elec. & Mfg. Co., Buffalo, N. Y.	J. J. Landers A. L. Stevens	J. S. Snyder Joseph Davis
HUGGINS, L. G., Asst. Mgr., Air Cond. Dept., Westinghouse Elec. & Mfg. Co., Springfield, Mass.	T. H. Urdahl R. E. Holmes	A. H. Goelz P. J. Marschall
JACKSON, W. F., Lab. Engr., American Stove Co., Lorain, Ohio.	D. L. Taze G. L. Tuve	C. A. McKeeman R. T. Southmayd
LYFORD, R. G., Sales Engr., The Powers Regulator Co., Dallas, Texas	I. I. Bock Herman Blum, Jr.	A. M. Harris R. K. Werner
MANNEN, D. E., Jr., Vice-Pres., The Mannen & Roth Co., Cleveland, Ohio	L. T. Avery L. E. Slawson	J. E. Wilhelm C. M. H. Kaercher
MCCLANAHAN, L. C., Dist. Mgr., Aerofin Corp., Dallas, Texas (*Reinstatement*)	W. H. Moler C. L. Kribs, Jr.	O. H. Mehl R. F. Taylor
MCGOWN, F. H., Jr., Student, Yale University, New Haven, Conn.	L. E. Seeley G. S. Williams	E. J. Rodee F. W. Keator (*ASME*)
MILLS, R. H., Sales Engr., W. S. Turner & Co., Portland, Ore.	J. D. Kroeker T. E. Taylor	C. E. Heinkel B. W. Farnes
NICOLL, S. F., Air Cond. Engr., York Ice Machinery Corp., York, Pa.	W. E. Zieber J. R. Hertzler	W. E. Barnum, Jr. R. F. Muhleman
NORFOLK, L. W., Consulting Engr. & Partner, E. G. Phillips Son & Norfolk, Nottingham, England.	G. N. Haden Walter Yates	E. G. Phillips (*IHVE*) John Hughes (*IHVE*)
STEVENS, W. H., Engr., Shellenberger, Gregg & Co., Milwaukee, Wis.	C. H. Randolph Ernest Szekely	H. C. Frentzel T. M. Hughey
SWEENEY, R. H., Sales, Minneapolis-Honeywell Regulator Co., Pittsburgh, Pa.	F. C. Houghten K. C. Lewis	F. B. Mahon J. F. S. Collins, Jr.
WOODGER, H. W., Maintenance Engr., General Elec. Co., Pittsfield, Mass.	H. J. Ryan C. H. Quirk	G. A. Teeling E. G. Atkinson (*AIA*)
YATES, J. E., Asst. Engr., Pacific Power & Light Co., Portland, Ore.	J. D. Kroeker J. A. Freeman	B. W. Farnes C. E. Heinkel

CANDIDATES ELECTED

In the past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

ADAMS, C. Z., Br. Mgr., Ilg Electric Ventilating Co., Greensboro, N. C.

ADEMA, G. E., Htg. Contractor, N. M. Adema & Son, Buffalo, N. Y.

ARMSTRONG, C. E., Chief Engr., Armstrong Heat Control Co., Portland, Ore.

BARNES, R. W., Htg., Vtg. & Air Cond. Contractor, San Antonio, Texas

BURKE, J. J., Engr., Carrier Engrg. Corp., Philadelphia, Pa. (*Reinstatement*)

CLAY, WHARTON, Secy., National Mineral Wool Assn., New York, N. Y. (*Advancement*)

COMO, J. A., Engr., Buckhead Plbg. & Htg. Co., Atlanta, Ga.

COX, S. F., Dir. Tech. Sales, Pittsburgh Corning Corp., Pittsburgh, Pa.

DILL, R. S., Assoc. Mech. Engr., National Bureau of Standards, Washington, D. C.

HACH, E. C., Chief Engr., Sales Dept., Standard Air Cond., Inc., New York, N. Y.

HAWK, J. K., Engr., General Air Cond. Co., Buffalo, N. Y. (*Advancement*)

KEANE, G. F., In charge of Engrg., Cooney Refrigeration Co., Syracuse, N. Y.

KOLB, R. P., Prof. Heat Power Engrg., Worcester Polytechnic Inst., Worcester, Mass.

KOLLAS, W. J., Chief Engr., Montag Stove & Furnace Works, Portland, Ore.

MABLEY, T. H., Chief Engr., Mechanical Heat & Cold, Inc., Detroit, Mich.

McINDOE, J. F., Mgr. Htg. Dept., Consolidated Supply Co., Portland, Ore. (*Reinstatement & Advancement*)

OERTEL, F. H. E., Engr. & Field Repr., H. C. Baker Co., Inc., Greensboro, N. C.

ORGAN, FREDERICK, Archt. & Engr., Omaha, Nebr.

PETIGROW, B. N., Sales Engr., Carrier Corp., Tel Aviv, Palestine.

POUGHER, E. W., Director, Messrs. Ernest W. Pougher & Son, Manchester, England.

SABLE, E. J., Treas., The T. O. Murphy Co., Oberlin, Ohio.

STIRLING, W. N., Tech. Repr. for India, Messrs. J. & E. Hall, Ltd., Dartford, Kent, England.

SWAIN, W. L., Director, Young, Austen & Young, Ltd., London, W. C. 2, England

TAHRY, M. EL, Engr. in Charge of Design, Carrier Egypt, S. A. E., Cairo, Egypt.

WILLIAMS, L. G., Engr., Williams Plbg. Co., Gladstone, Ore.

ASSOCIATES

ANDERSON, E. J., Mfrs. Agent, Detroit, Mich.

BLAKER, A. H., Secy-Treas., National Korectaire Co., Chicago, Ill.

BREX, I. E., Asst. Secy. & Treas., Brex & Bieler, Inc., Brooklyn, N. Y.

CLARK, A. C., Engr., Century Co., Portland, Ore.

DE BERARD, P. E., Pres., Conditioned Air Systems, Inc., Chicago, Ill.

FOX, W. K., Supt., Northwest Stove Works, Inc., Portland, Ore.

GOEHLER, E. E., Pres., Vortex Mfg. Co., Portland, Ore.

ISAACS, H. A., JR., Pres., Oil Delivery, Inc., Red Bank, N. J.

JOLLEY, E. M., Sales Dept., The Marley Co., Kansas City, Kans.

KLIEFOTH, M. H., Treas., Research Products Corp., Madison, Wis.

LA RUE, J. A. D., Mgr., Roofers, Inc., Montreal, Que., Can.

MACGREGOR, C. M., Sales Engr., Portland Gas & Coke Co., Portland, Ore.

MATTINGLY, M. F., Sales Engr., Johnson Service Co., Chicago, Ill.

MEAD, H. K., Mfrs. Agent, Portland, Ore.

MICHAELS, M. A., Owner, Mgr., Century Co., Portland, Ore.

MOORE, B. W., Mgr., Htg. Dept., F. B. Connelly Co., Portland, Ore.

MUHLEMAN, R. F., Asst. Mgr., York Ice Machinery Corp., York, Pa.

NIELSEN, H. B., Mfrs. Repr., Portland, Ore.

PREWITT, H. B., Sales Engr., American Blower Corp., Philadelphia, Pa.

SIEGEL, R. C., Owner, International Chimney Co., Buffalo, N. Y.

STEWART, J. N., Smoke Inspector, Div. of Smoke Regulation, Washington, D. C.

TURNER, E. S., Sales Engr., Wm. S. Turner & Co., Portland, Ore.

URBAN, F. F., Design & Estimating, Urban Plbg. & Htg. Co., Portland, Ore.

WALZ, C. D., Assoc. Mech. Engr., 11th Naval Dist., San Diego, Calif.

WELLER, A. K., Factory Agency, Portland, Ore.

WILLIAMS, E. C., Mfrs. Repr., Cincinnati, Ohio.

JUNIORS

BAILEY, C. F., Instructor, Mech. Engrg., Mississippi State College, State College, Miss.

BANACH, C. J., Chief Draftsman, Johnson Fan & Blower Corp., Chicago, Ill.

BIERINGER, F. A., Asst. Chief Househeating Engr., Brooklyn Borough Gas Co., Brooklyn, N. Y.

BORTON, A. R., Sales Engr., John J. Nesbitt, Inc., Pittsburgh, Pa.

EMANUELS, MASON, Sales Engr., Pacific Scientific Co., San Francisco, Calif.

GOLDSMITH, ELLIOT, Design Engr., Anemostat Corp. of America, New York, N. Y.

HANTHORN, WALTER, Engr., Kleenair Furnace Co., Portland, Ore.

HILL, EDWARD, JR., Htg. Engr., Fox Furnace Div., American Radiator Co., San Francisco, Calif.

HYDE, E. V., Sales, Wood Conversion Co., St. Paul, Minn.

LEVINE, CHARLES, Sales Engr., General Appliances Sales Co., Inc., Brooklyn, N. Y.

McCLUNG, T. H., Sales Engr., Brod & McClung, Portland, Ore.

MORRISON, W. B., Engr., Meier & Frank Co., Inc., Portland, Ore.

PELTZMAN, J. L., Htg. & Vtg. Engr., E. K. Campbell Htg. Co., Kansas City, Mo.

PETERSON, H. P., Air Cond. Engr., Bush Mfg. Co., Hartford, Conn.

WALDREP, J. E., Air Cond. Engr., Poe Piping & Heating Co., Greenville, S. C.

WILDMAN, E. L., Air Cond. & Design Engr., Stewart A. Jellett Co., Philadelphia, Pa.

WILLIAMS, H. E., Jr. Engr., Nash-Kelvinator Corp., Long Island City, N. Y.

STUDENTS

BACHMAN, A. J., Student, Pratt Inst., Brooklyn, N. Y.

BATES, J. H., Student, Purdue University, West Lafayette, Ind.

BRAUN, C. R., Jr., Student, Carnegie Inst. of Tech., Pittsburgh, Pa.

BUCKLEY, D. J., Student, University of Illinois, Urbana, Ill.

FREEMAN, A. W., Student, New York Tech. Inst., New York, N. Y.

HAMPLE, H. J., Student, Carnegie Inst. of Tech., Pittsburgh, Pa.

LEITGABEL, K. A., Student, University of Wisconsin, Madison, Wis.

MUMFORD, W. W., Machinist's Mate, U. S. Navy, New York, N. Y.

WIESNER, B. K., Student, Purdue University, West Lafayette, Ind.